NASA CR 51675

167 p.

[redacted]

ARMY BALLISTIC MISSILE AGENCY

REDSTONE ARSENAL, ALABAMA

ABMA Form 485-1, 1 Apr 59

(NASA CR-51675;

/ABMA

17 December 1959 Report No. DLS-TN-26-59

17

167 p ref

cy HI-1

LUNAR SOFT LANDING STUDY
FOR
NATIONAL AERONAUTICAL AND SPACE ADMINISTRATION

SYSTEMS SUPPORT EQUIPMENT (U)

Compiled by Owen L. Sparks comp. 1

[redacted]

0787002

2/ SYSTEMS SUPPORT EQUIPMENT LABORATORY
ARMY BALLISTIC MISSILE AGENCY

, Huntsville, Ala.

PREFACE

SECOND EDITION

The first edition of this study was given only the limited distribution necessary to facilitate preparation of a consolidated ABMA report to NASA. This second edition was printed to meet the requirement for wider distribution to the ABMA laboratories concerned with the soft lunar landing equipment problems. This edition is essentially identical to the first edition with the exception of minor corrections and the substitution of more up-to-date illustrations of certain ground support equipment as indicated below.

New Figures Substituted in This Edition

Figure 10. Lower Booster Firing and Service Equipment Installation

Figure 11. Upper Booster Firing and Service Equipment Installation

Figure 16. LOX Storage Area - SATURN

Figure 17. LOX Storage and Transfer System (SATURN)

Figure 19. Fuel System Schematic (SATURN)

FOREWORD

The information contained in this document is intended primarily for use of ABMA in the preparation of a "Lunar Soft Landing Study" for the National Aeronautical and Space Administration. However, it may be of interest to other personnel concerned with the development of the SATURN system and with other space projects.

Material in this document is presented in three section: Section I, Ground Support Equipment; Section II, Manned Lunar Capsule Recovery; and Section III, Lunar Roving Vehicle. It should be pointed out that a majority of the ground support equipment described in Section I has already been developed or has been designed for use in the overall SATURN Vehicle Program. Therefore, most of this section is a proposal only in the sense that it is proposed to use existing SATURN Ground Support Equipment in the Lunar Soft Landing System. However, the information presented in Section II and Section III is based on preliminary design studies only and is submitted as a proposal subject to a complete R&D Program.

Since information presented herein is very general in nature it is not intended for use as final design criteria.

TABLE OF CONTENTS

SECTION I. GROUND SUPPORT EQUIPMENT

TABLE OF CONTENTS (CONT'D)

TABLE OF CONTENTS (CONT'D)

TABLE OF CONTENTS (CONT'D)

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (CONT'D)

LIST OF ILLUSTRATIONS (CONT'D)

LIST OF TABLES

LIST OF APPENDICES

SECTION I

GROUND SUPPORT EQUIPMENT

A. LOGISTICAL CONSIDERATIONS

1. General

Logistical support of the launching of lunar mission vehicles from the Atlantic Missile Range will present major transportation problems. The four major missile components (booster, second and third stages, and payload) must be shipped to AMR from different sections of the country and all of these components have dimensions which exceed the maximum capabilities of conventional air, rail or road carriers. Special equipment and special routing must be provided and the overall program schedule will not allow sufficient time for movement of these items at the most convenient times consistent with other transportation activities. Several requirements will be discussed in the following paragraphs to illustrate the problems involved.

2. Booster

The SATURN Booster to be transported from ABMA to AMR in an assembled condition will be 256 inches in diameter and approximately 82 feet in length. Since the booster will be moved on its transporter, these dimensions will become even larger. Therefore, the only way to transport such a large, expensive and vulnerable item from ABMA to AMR is by waterway. A rather detailed description of the facilities and equipment for handling and transporting the booster over this route is given in paragraph B. 1. below. The barge transportation described will assure safe handling of the cargo but will require between two and three weeks for the transportation phase alone. It will be necessary to

partially build and/or reinforce about 10 miles of roadway at Redstone Arsenal and about two miles of roadway at AMR. Obstacles such as power and telephone lines must be removed, and for the actual movement, all other road traffic must be blocked or rerouted. Since the barge must be returned to RSA with the recovered booster and maintenance time for barge and equipment must be allowed, it is obvious that firings at two-month intervals will require at least two barges and personnel for continuous operation.

3. Second and Third Stages

Since a decision has not been reached at this time on the diameter of the upper stages of the vehicle to be used for this mission, both possibilities will be briefly discussed.

a. 120 Inches Diameter. Air transportation of components of this diameter is possible with the C-133 aircraft which has cargo compartment dimensions of 140 inches wide, 144 inches high and 85 feet long. This is adequate for either of the upper stages. However, this is the only aircraft presently available with this capability. With air transportation available (it is assumed that adequate aircraft of this type will be available for this purpose by 1963), there should be no major problems involved in scheduling deliveries of upper stages of this diameter to meet tight fabrication and firing schedules.

b. 160 Inches Diameter. Transportation of stages of this diameter over long distances from an inland area such as Denver, Colorado to AMR presents a critical logistical problem. Air transportation by the largest presently available cargo aircraft (C-133) is not possible. Water transportation in a manner similar to that used for the booster would

be the most simple method but is not always possible. Road transportation is possible but will require special routing and traffic restriction. Rail transportation would be even more difficult. Therefore, transportation by a lighter-than-air craft (blimp) has been considered but may prove to be economically impractical. No conclusion has been reached on a solution to this problem at the present time.

4. Booster Recovery

Handling of the booster after a successful landing and recovery also constitutes a major operation. The operation of spotting and water recovery of re-entry nose cones is well known. However, the water recovery of a small and compact nose cone is far more simple than that of the voluminous, delicate booster which is partially filled with propellant residuals, and therefore, presents a safety hazard to equipment and personnel. Details of the proposed water recovery scheme are presented in paragraph D. A Landing Ship Dock (LSD) is proposed as the main equipment. Flushing, inspection, preservation and disassembly of delicate parts will be done on board the ship by a special crew during the return trip. Later the missile will be transferred to a river barge for return to RSA dock. A suitable harbor with proper crane facilities, such as New Orleans, will be used for this purpose. Preparation, recovery action and return to New Orleans will require an LSD for about one week and supporting ships for a portion of this time.

B. TRANSPORTATION TO LAUNCH SITE

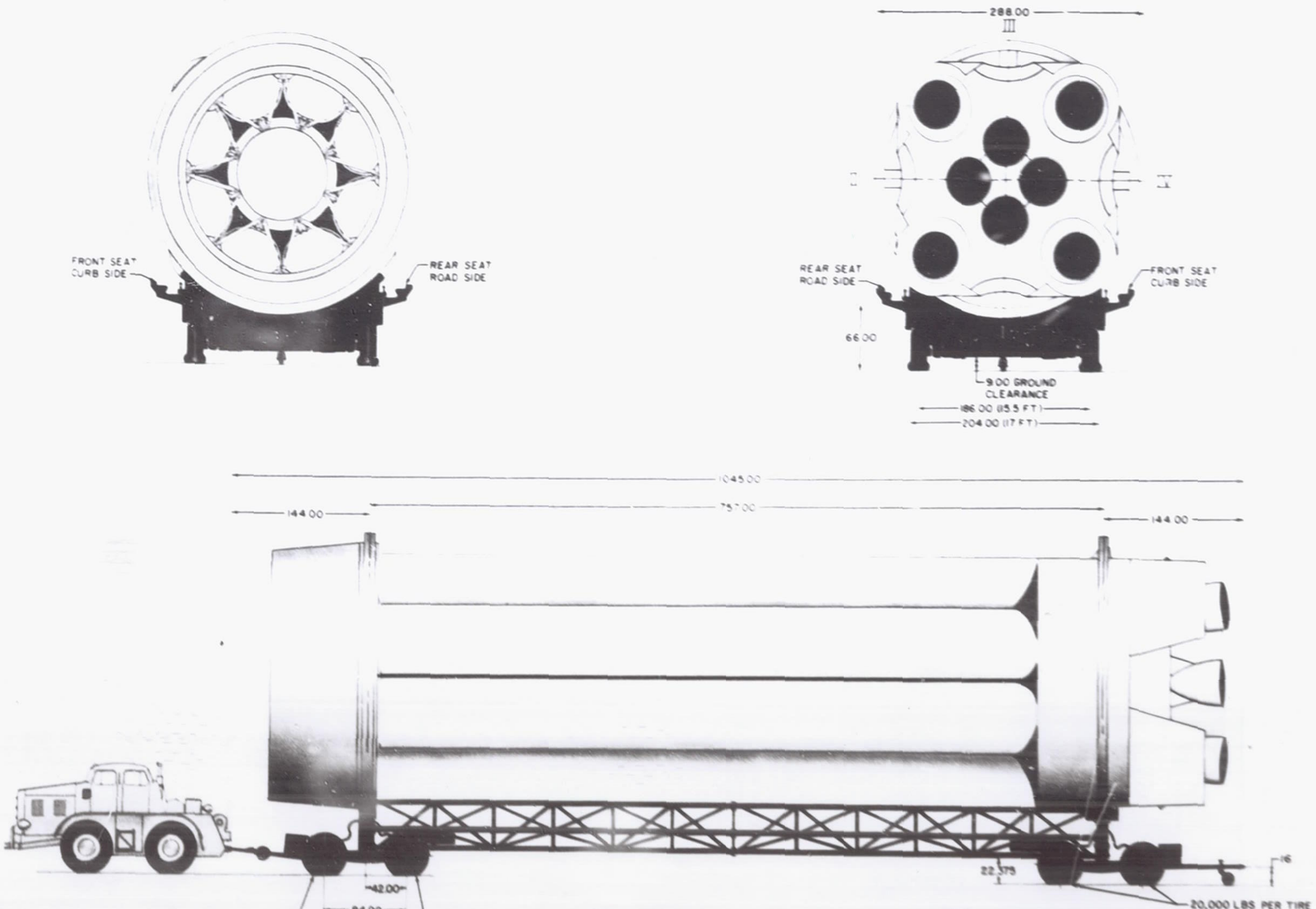
1. Booster

a. Land Transportation (Figure 1) (Figure 2)

(1) Transporter. (Figure 1). The booster transporter is a unique piece of equipment in that a part of the booster final assembly jig is used to make up the transporter assembly. The assembled booster, with its Support Cradles, Connecting Trusses and Assembly Rings, is jacked up as a unit and placed on two axle and wheel assemblies. Pertinent transporter dimensions and wheel and axle loads are shown on the silhouette representation of the missile-transporter combination. Each wheel assembly consists basically of two pairs of two independently braked and hydraulically steered aircraft tandem wheels on an axle assembly. The Support Cradles are secured to the axle assemblies and a towbar on the forward axle assembly connects to the prime mover. The booster is carried on this composite vehicle through all phases of testing, checkout, and transportation from the Fabrication Laboratory to the launch site.

(2) Prime Mover. (Figure 2). The prime mover for the transporter during all land transportation is a standard aircraft tug (TT-11). The maximum towed speed of the loaded transporter is between 3 and 5 mph. The TT-11 aircraft towing tractor is powered by a Buda diesel engine with a maximum draw bar pull of 13,500 pounds. The transmission consists of a 3 stage twin disc torque converter. The maximum angle of approach and departure is 13° and 17° respectively.

SATURN TRANSPORTER AND TOWING VEHICLE



NOTES:
1. TOWING VICHICLE - TRACTOR, AIRCRAFT TOWING COLEMAN
   TUG GROSS WT. 33500 LBS.
2. TIRES - 1700 X 16 TYPE III 10 PLY NYLON
   OUTSIDE DIA. 45"
   WIDTH 17.34
   ROLLING RADIUS @ 35% DEFLECTION 17.8"
   CONTACT AREA @ RATED LOAD OF 31% DEFLECTION 275 IN^2
   @ 35% DEFLECTION 300 IN^2 MAX
   TIRE PRESSURE - REAR 80 ± 10 PSI FRONT 60 $^{+30}_{-10}$ PSI

Figure 1. SATURN TRANSPORTER AND TOWING VEHICLE

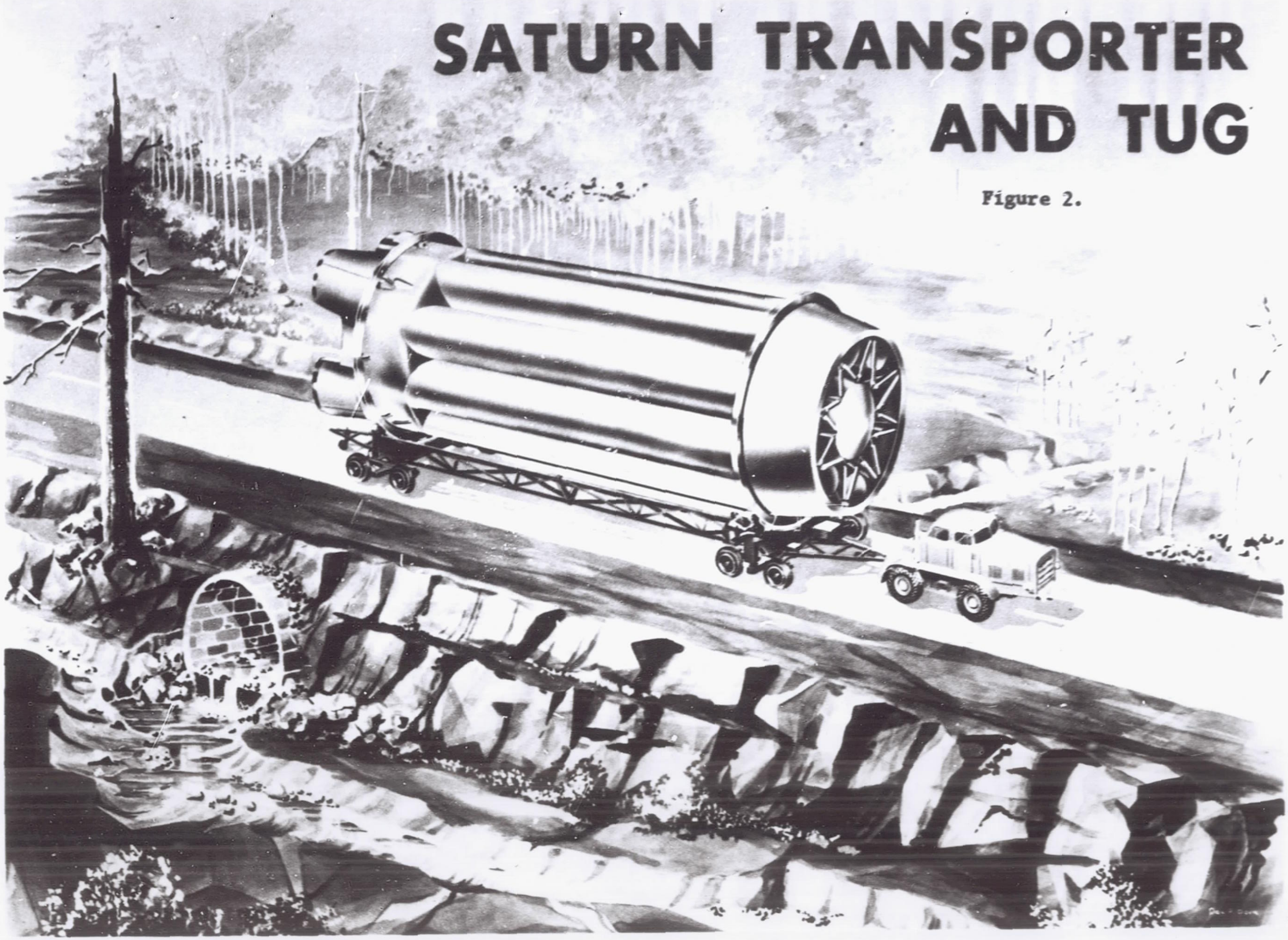


Figure 2.

b. Dockside Facilities

(1) Redstone Arsenal Dock (Figure 3). After all booster testing at ABMA is completed, the booster-transporter combination is towed to the RSA Dock and is rolled on to a specially designed barge. The dockside facilities for this operation consist of a ramp to the waters edge and two electrically powered winches mounted at the top of the ramp to control movement of the transporter up and down the ramp. An undamaged booster returned to RSA Dock on a transporter will be off-loaded in a similar manner. However, when a heavily damaged booster is returned it may not be supported on a transporter and partial salvage operations may be necessary before off-loading is accomplished. In this case lift facilities, such as a mobile crane, will be necessary for removing salvaged components from the barge. As the recovery program becomes more extensive and if the return of heavily damaged boosters requires heavier off-loading facilities, the RSA Dock will have the capability for expansion to include construction of a 100 ton stiff-leg derrick with a reach of 85 feet to provide lifting facilities for a complete booster when it is returned in a damaged condition without its transporter.

(2) New Orleans Port. Existing heavy lift facilities at the New Orleans Port are considered sufficient for transferring the recovered booster from the recovering LSD to the barge for return to RSA Dock.

(3) Atlantic Missile Range. Ramp facilities similar to those described for RSA Dock will be used for unloading the booster-transporter at Site C, AMR, approximately one mile from Complex 34 on the Banana River.

c. Water Transportation (Figure 4).

Two types of barges are presently being considered for the water transportation phase of booster movement from ABMA to AMR. The results of an investigation presently being carried out by the Transportation Corps in cooperation with Systems Support Equipment Laboratory will determine whether a coast-wise type barge or a full sea-going type barge will be used. Maximum booster protection, simplicity and cost of operation will be among the determining factors considered. Both barges will have end opening doors for roll-on/roll-off operations and removable hatch covers to provide complete environmental and hazard protection to the booster during transport. Operations with each type barge are discussed below.

(1) Coast-Wise Barge. With this type barge the booster-transporter is loaded on the barge at RSA Dock using the roll-on/roll-off operation described in paragraph B.1.b(1) above. The barge is towed down the Tennessee, Ohio and Mississippi Rivers to the New Orleans port by river tug. At New Orleans the entire barge is floated into a sea-going landing ship dock (LSD) which transports the cargo from New Orleans around the Florida Peninsula to Fort Pierce, Florida, approximately 80 miles south of Cape Canaveral. There the barge is discharged from the LSD and towed by river tug up the Indian River and Banana River to the barge basin at Site C, AMR, where the booster-transporter is off-loaded and towed to the SATURN Launch Site Complex.

(2) Sea-Going Barge. With this type barge the loading at RSA Dock would also be accomplished by the roll-on/roll-off method.

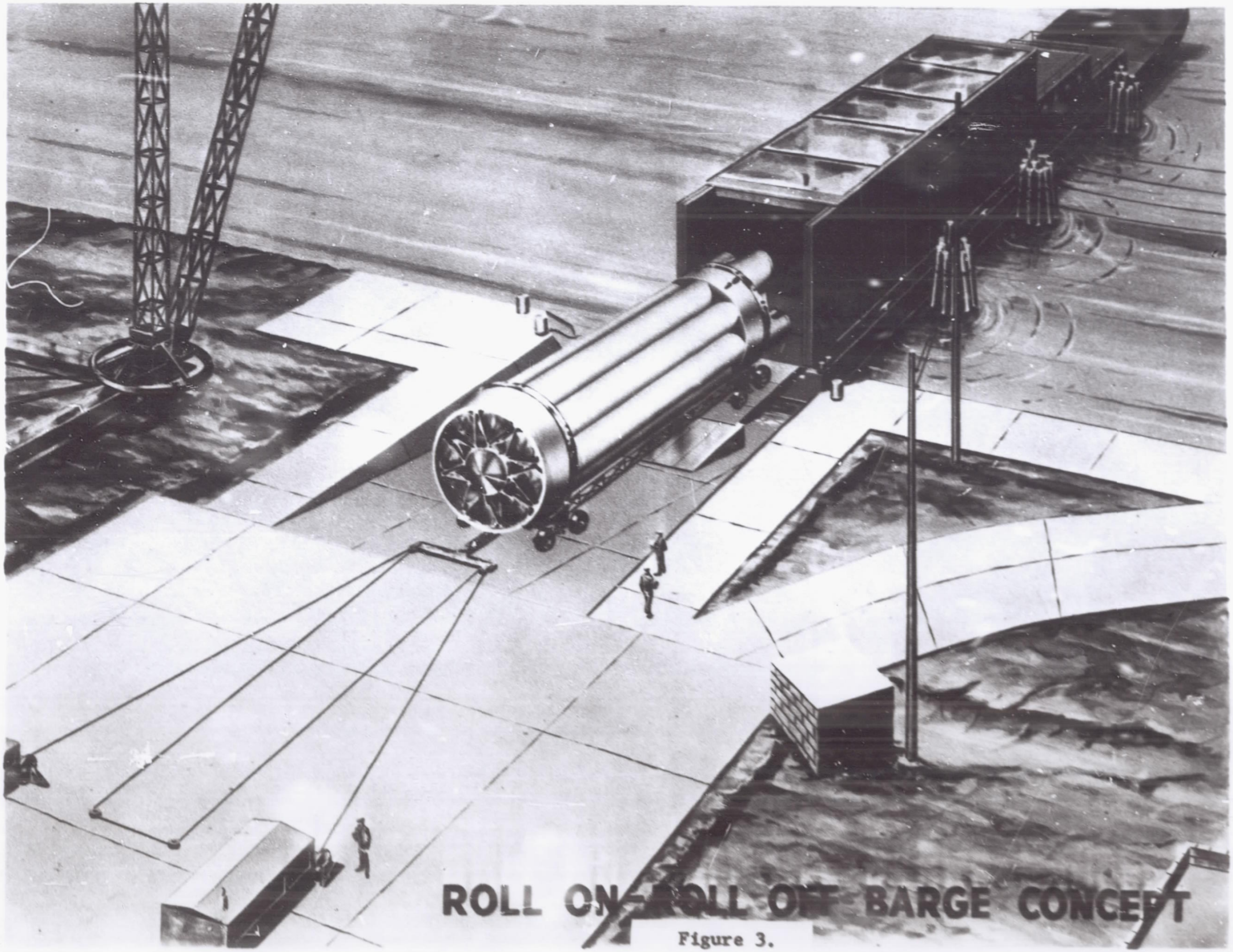


Figure 3.

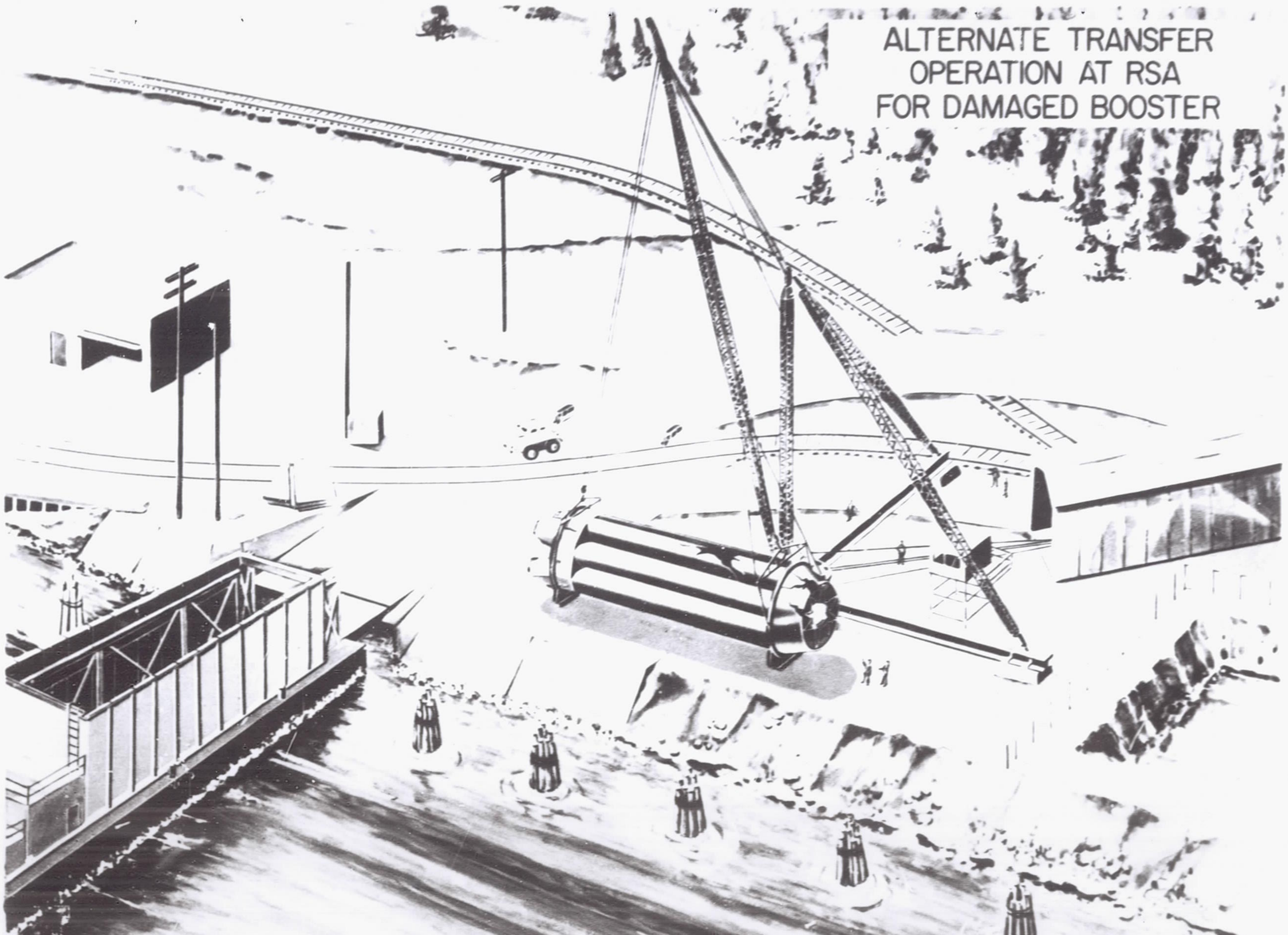


Figure 3a.

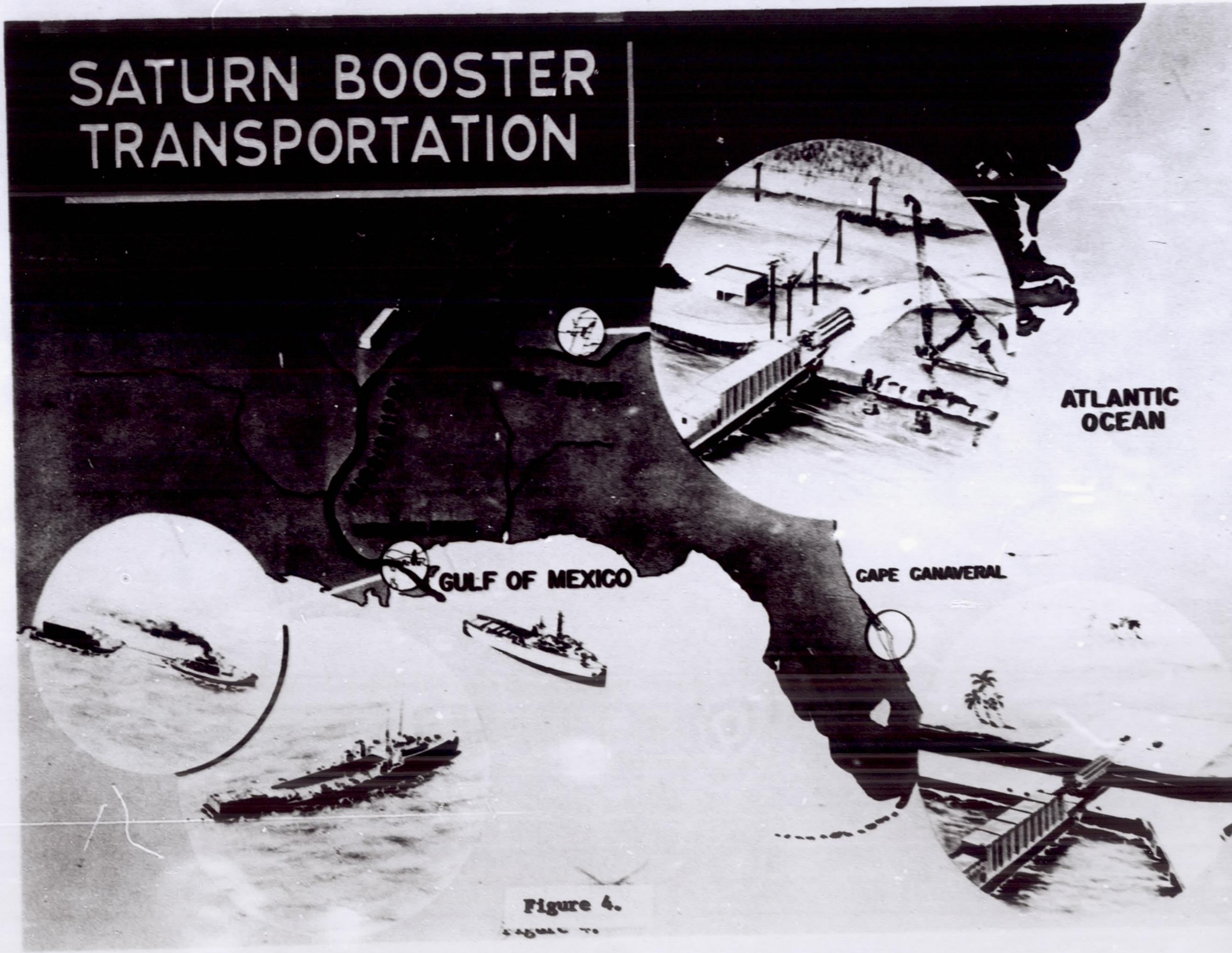

SATURN BOOSTER
TRANSPORTATION
ATLANTIC
OCEAN
GULF OF MEXICO
CAPE CANAVERAL

Figure 4.

14

At New Orleans the river tug (approximately 65 feet long) would be exchanged for a sea-going tug (approximately 100 feet long) which would then tow the barge to Fort Pierce, Florida. Here the sea-going tug would be exchanged for a river tug which would complete the trip to Site C barge basin. The obvious advantage of this method of transportation is that no LSD loading and unloading of the barge is required with the inherent possibility of damage to booster.

2. <u>Upper Stages</u>

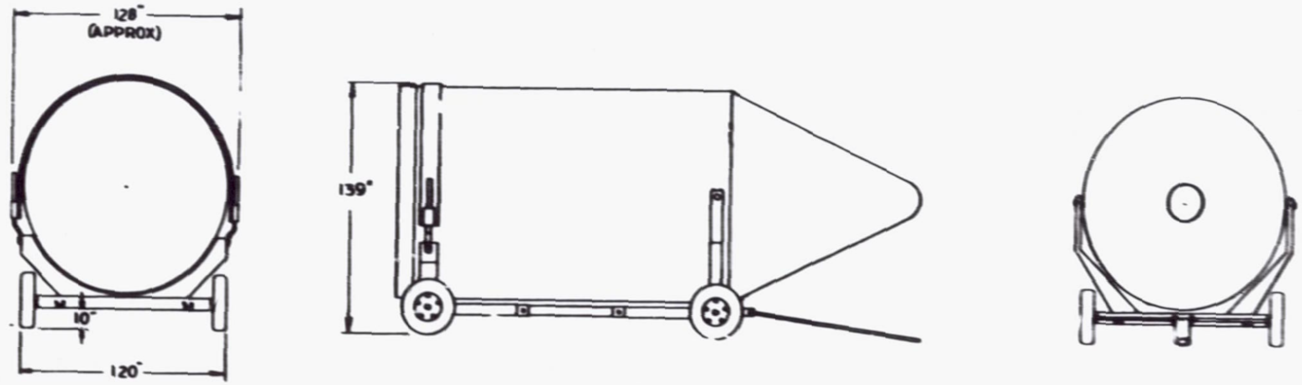
The second and third stages, to be fabricated, assembled and tested by Martin-Denver and Convair Astronautics respectively, may present a considerable transportation problem depending upon the final decision on the diameter of these stages. As pointed out in paragraph A.3. air transportation of a 120 inch diameter stage can be accomplished with the C-133 aircraft, whereas, no presently available aircraft can transport a 160 inch diameter stage. Since the final decision on the diameter of these stages has not been made no proposal on transportation will be made in this document. Details on handling equipment are also unavailable at this time.

3. <u>Payload</u>

a. <u>Transporter</u>. (Figure 5). The proposed transporter for the lunar payload is essentially a modified version of a commercially available four wheel transporter. The payload rests on a rear saddle and is restrained and supported at the forward end by its lifting bolts. This is a proven system for missile transporters and it effectively prevents torsional strains from being introduced into the missile. Overall dimensions are shown on the silhouette illustration of the transporter with payload.

b. <u>Prime Mover</u>. Any commercial type small tug tractor with pintle hook may be used as a prime mover for the payload transporter since road movement will normally be limited and will be accomplished on first class roads. However, road speeds must not exceed five miles per hour to prevent shock loading of the payload in excess of 4g's.

DD FORM 1176-3-R



ORDNANCE PART NO.	
ORIGINAL DATE OF DRAWING	4 SEPT 1959
DRAFTSMAN	CEB
SEE ENGINEERING RECORDS	
APPLICATION	
MATERIAL	
HEAT TREATMENT	
FINAL PROTECTIVE FINISH	
SCALE	

PAYLOAD
ON
TRANSPORTER

Figure 5.

ORDNANCE CORPS
DEPT OF THE ARMY
ARMY BALLISTIC MISSILE AGENCY

B

16

c. Transportation. (Figure 6). Payload dimensions will permit transport by C-133 aircraft and this will be the normal method of transportation to the launch site complex. Shipment of the payload will be made on the transporter as shown in the silhouette illustration to facilitate loading, unloading and tie-down in the aircraft. Either rail or road facilities may be used for return shipment of the reuseable materials and equipment from the AMR to ABMA.

d. Packaging. The outer skin of the payload will constitute the external container in which the internal components will be living in a controlled environment. End covers and strippable coating will complete the sealing requirements. Desiccant breathers and appropriate venting will provide protection and prevent excessive differential pressures during air transport. Environmental protection, will be provided by a water proof tarpaulin or insulated blanket as required by ambient conditions.

C. LAUNCH SITE

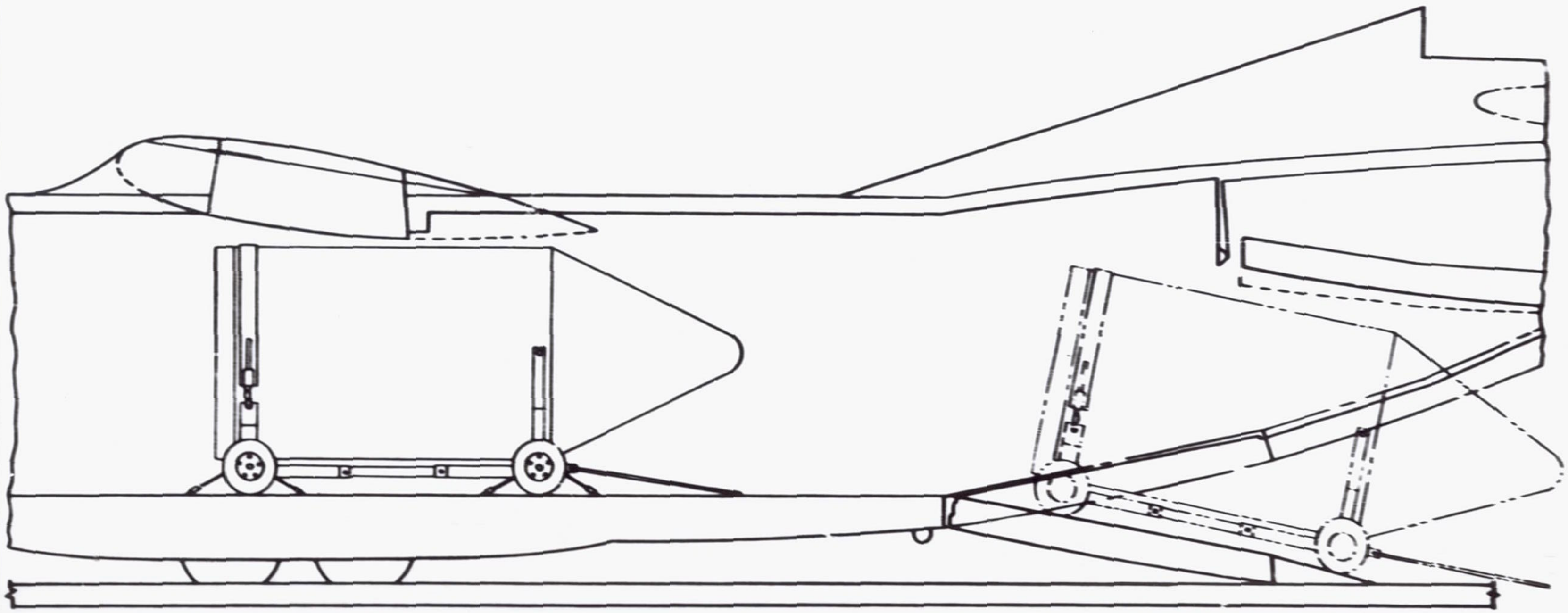
1. <u>Launch Site Complex</u> (Figure 7).

The SATURN Launch Complex contains all necessary facilities for handling, storing, servicing, checkout, erection and launching of the SATURN vehicle as well as the required administration and logistical facilities and special research laboratory facilities to support the various projects to be carried out during the SATURN Program.

Upon arrival at the barge basin the booster and transporter are off-loaded from the barge and towed to the Booster Assembly Building, where final assembly details and horizontal checkout are accomplished. From here the booster passes through the staging building to the launch pad for erection. The upper stage assemblies are handled in a similar manner from the airfield to launcher.

One launch pad facility is sufficient to support SATURN firings at approximately two-month intervals. Therefore, based upon the presently projected firing schedule, one pad facility will be sufficient to support all the Lunar Payload Missions. However, the propellant storage and transfer facilities are designed with the capability of supporting two launch pads at alternate intervals when required. This will be a requirement when the Lunar Payload Missions are superimposed on other SATURN program schedules.

1176-3-R



ORDNANCE PART NO.

SEE ENGINEERING RECORDS

APPLICATION

DO NOT APPLY PART NO. AS SPECIFIED

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES

MATERIAL

HEAT TREATMENT

FINAL PROTECTIVE FINISH

ORIGINAL DATE OF DRAWING 4 SEPT 1959

DRAFTSMAN CEB

APPROVED BY ORDER OF THE CHIEF OF ORDNANCE

LOADING DIAGRAM
PAYLOAD
IN C-133 AIRCRAFT

Figure 6.

SCALE

UNIT WT

ORDNANCE CORPS

DEPT OF THE ARMY

ARMY BALLISTIC MISSILE AGENCY

B

19

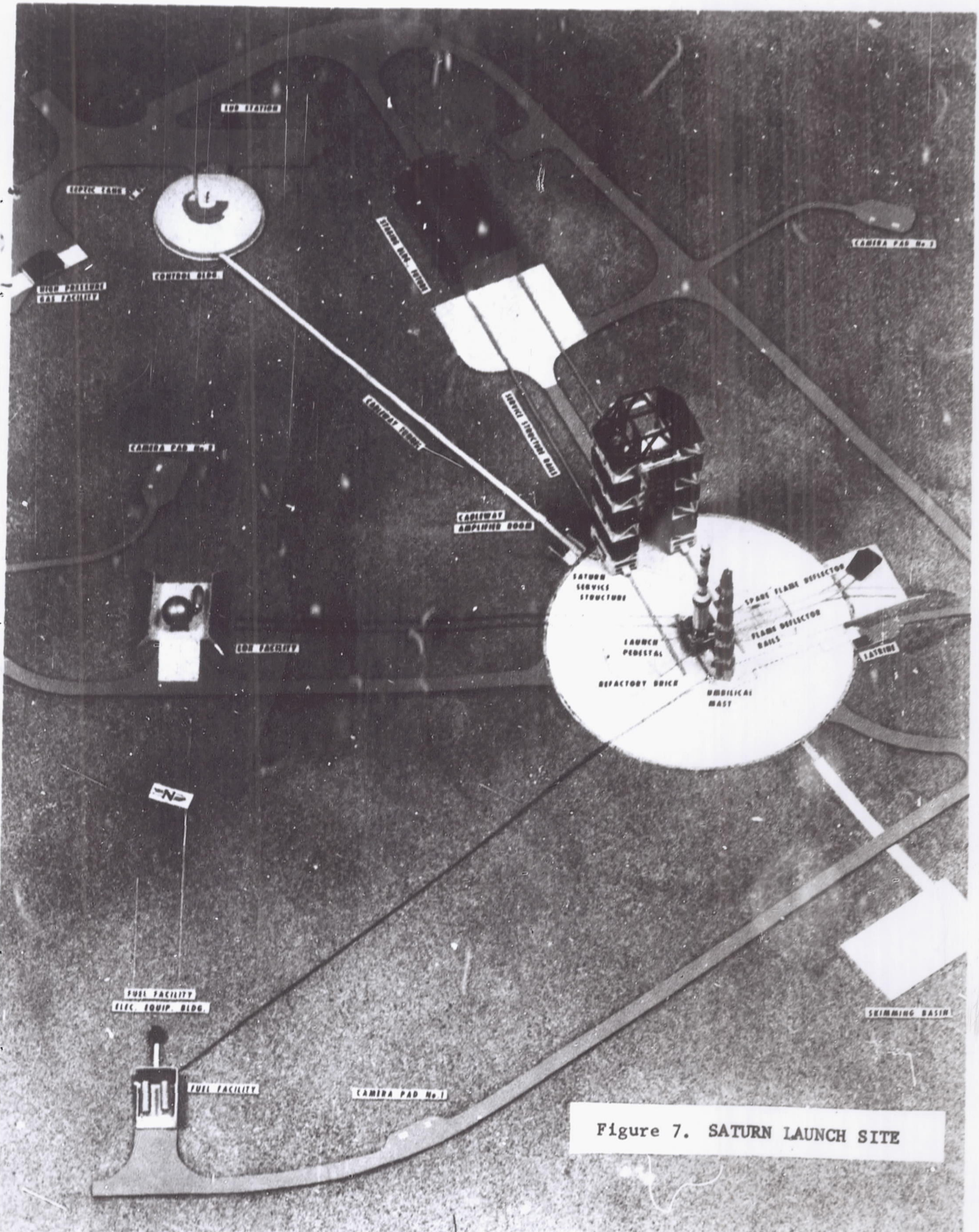


Figure 7. SATURN LAUNCH SITE

2. <u>Booster Erection</u> (Figure 8) (Figure 9)

The SATURN Booster is erected on the launcher by utilizing the track mounted gantry type service structure. This structure has a bridge crane supporting two hooks of 40 and 60 tons capacity each. The hooks are approximately 12 feet apart horizontally and can be moved horizontally, longitudinally and vertically. In preparation for erection the service structure is positioned over the launcher, the booster transporter is towed into position parallel to the service structure base, and the booster is rotated 45° from transporting plane to the erecting plane. The top portion of the rear assembly ring is removed from the booster and two tension cables are placed between the front hoist points and the rear thrust frame to minimize eccentric erection loads. Catenary effects in these cables are minimized by use of several connecting bands across the upper half of the booster. The gantry crane is then moved into position and connected to the booster pick up points by means of erection slings and beams. The 60-ton hook is connected to the forward sling and the 40-ton hook to the thrust frame sling. The booster is lifted from the transporter, rotated into vertical position, moved into the gantry structure and lowered onto the four preleveled launcher support and hold-down points with the assistance of removable guides attached to the launcher arms.

3. Booster Checkout (Figure 10) (Figure 11)

a. Leak and Function Checkout

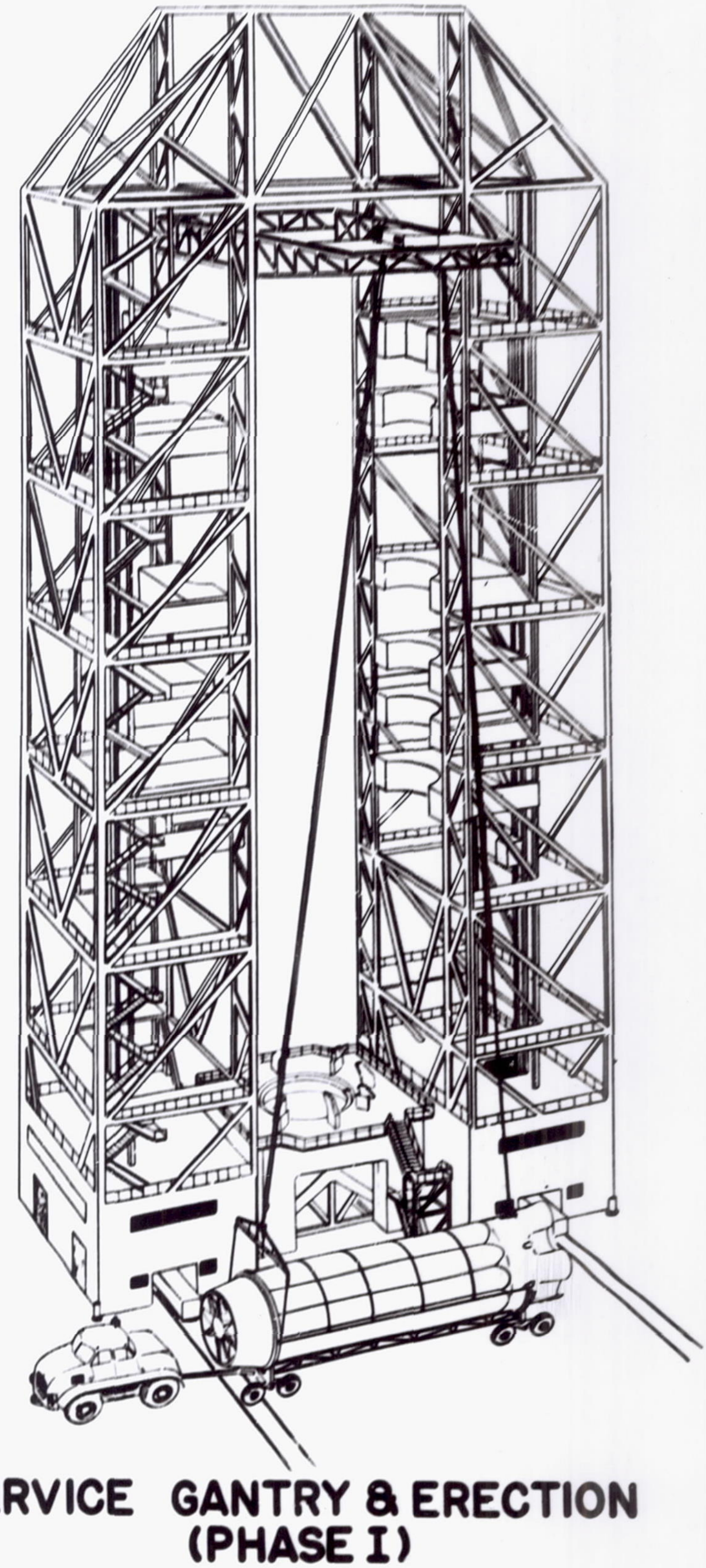
After erection, a pneumatic leak and function check is made on the booster to determine if any components or subsystems were damaged during transportation and handling. Gaseous nitrogen (GN_2), used in performance of the checks, is routed to the checkout panels on the launcher through a pneumatic distribution station in the base of the umbilical tower. The GN_2 pressure is regulated at these panels and, in conjunction with electrical control panels in the blockhouse, serve to distribute the GN_2 to various check-points throughout the system where function checks of the valves, leakage of joints and fittings, and flow rate checks are performed. Pneumatic distribution lines also extend from the pneumatic distribution station to checkout panels on the service structure for checkout of the top part of the booster.

b. Engine Servicing

Following pneumatic checkout, an engine service operation is performed using the Engine Servicing Trailer. This operation includes flushing and purging of the critical portions of the engine such as the fuel jacket, LOX dome, gas generator, and the fuel igniter line to insure absolute cleaniness of these parts.

4. Vertical Engine Removal

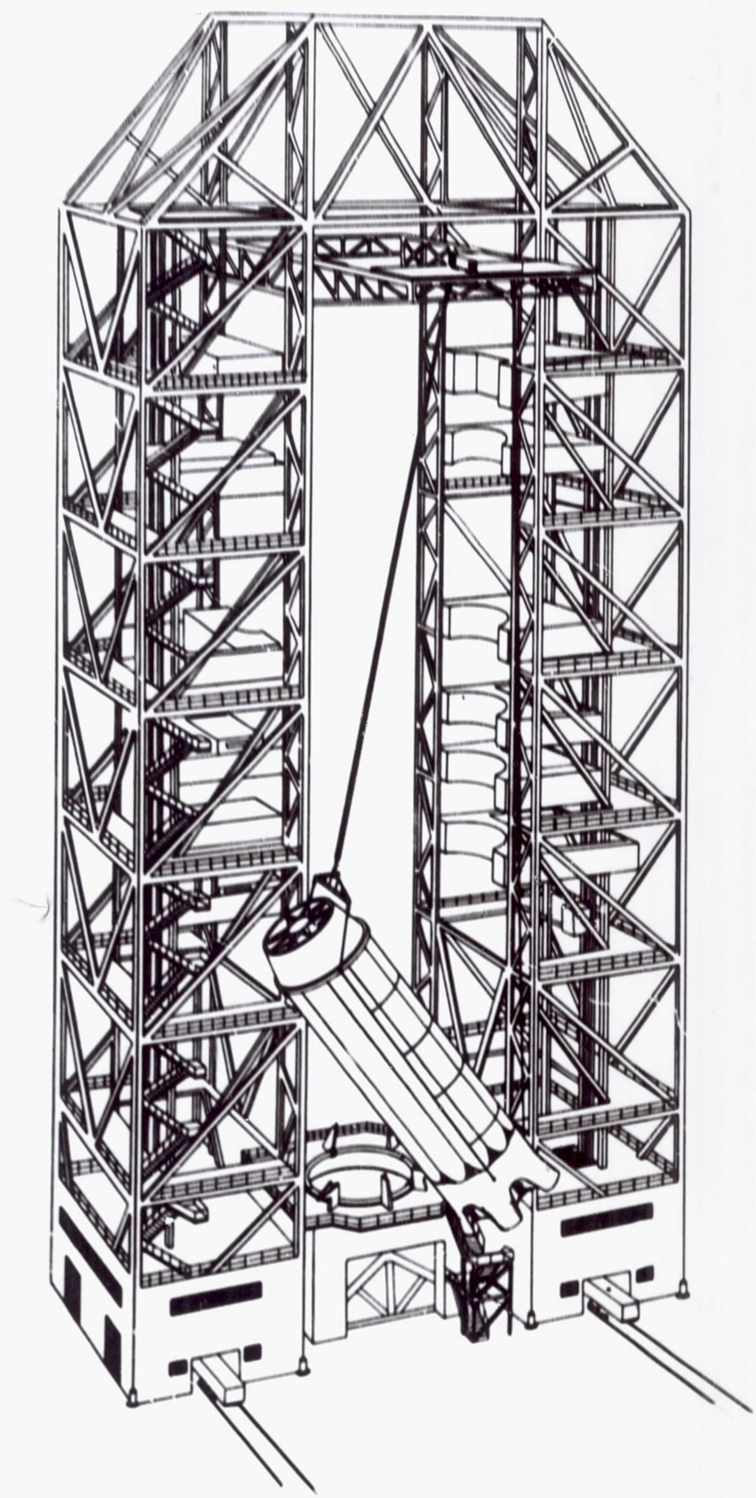
Lack of working space in the booster tail section area makes it necessary to have a handling and hoisting system capable of operation in a restricted area for removal and replacement of a defective engine after the booster is in the vertical position.



SERVICE GANTRY & ERECTION
(PHASE I)

Figure 8.

23



SERVICE GANTRY & ERECTION
(PHASE II)

Figure 9.

LOWER BOOSTER FIRING AND SERVICE EQUIPMENT INSTALLATION

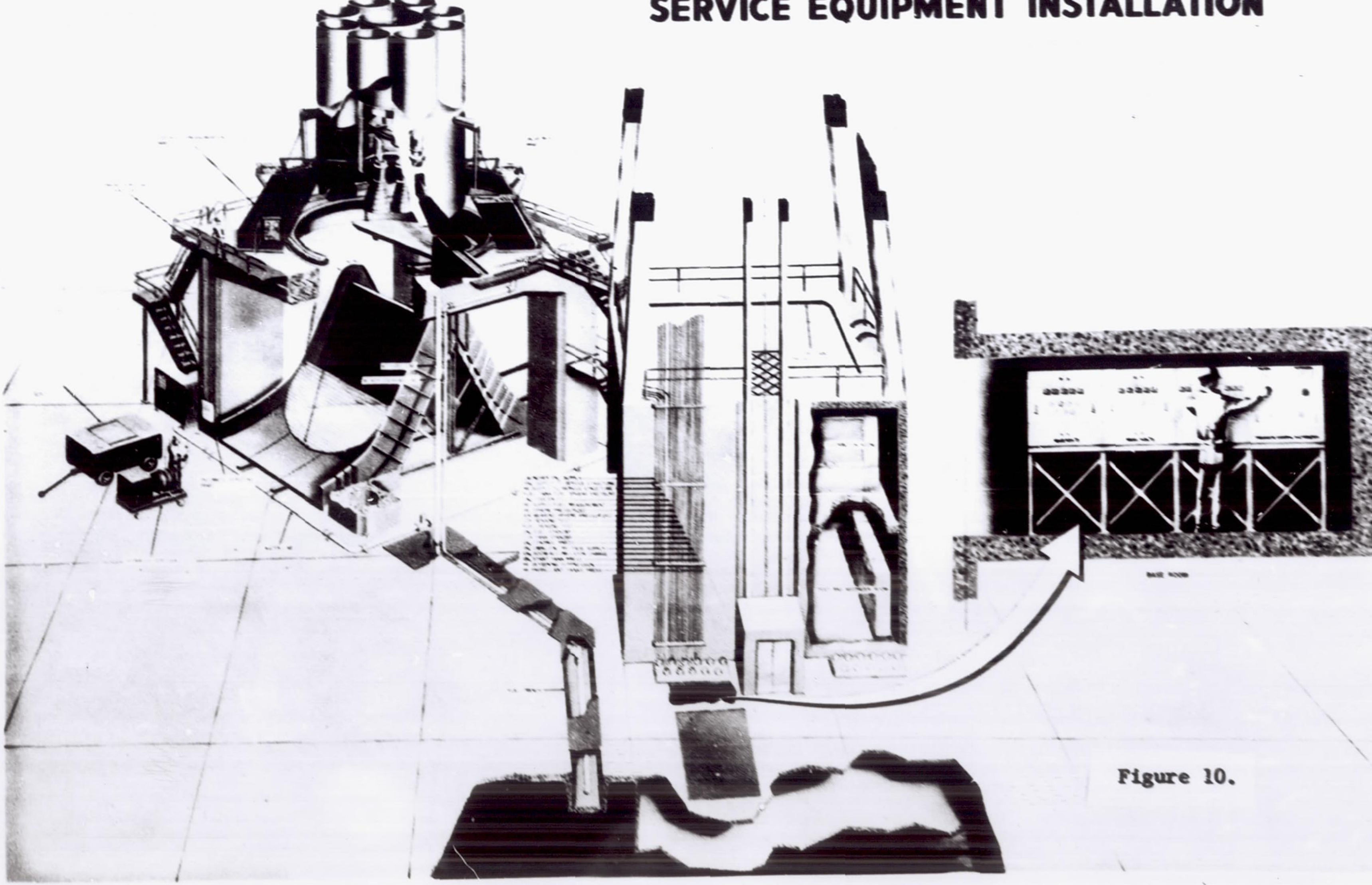
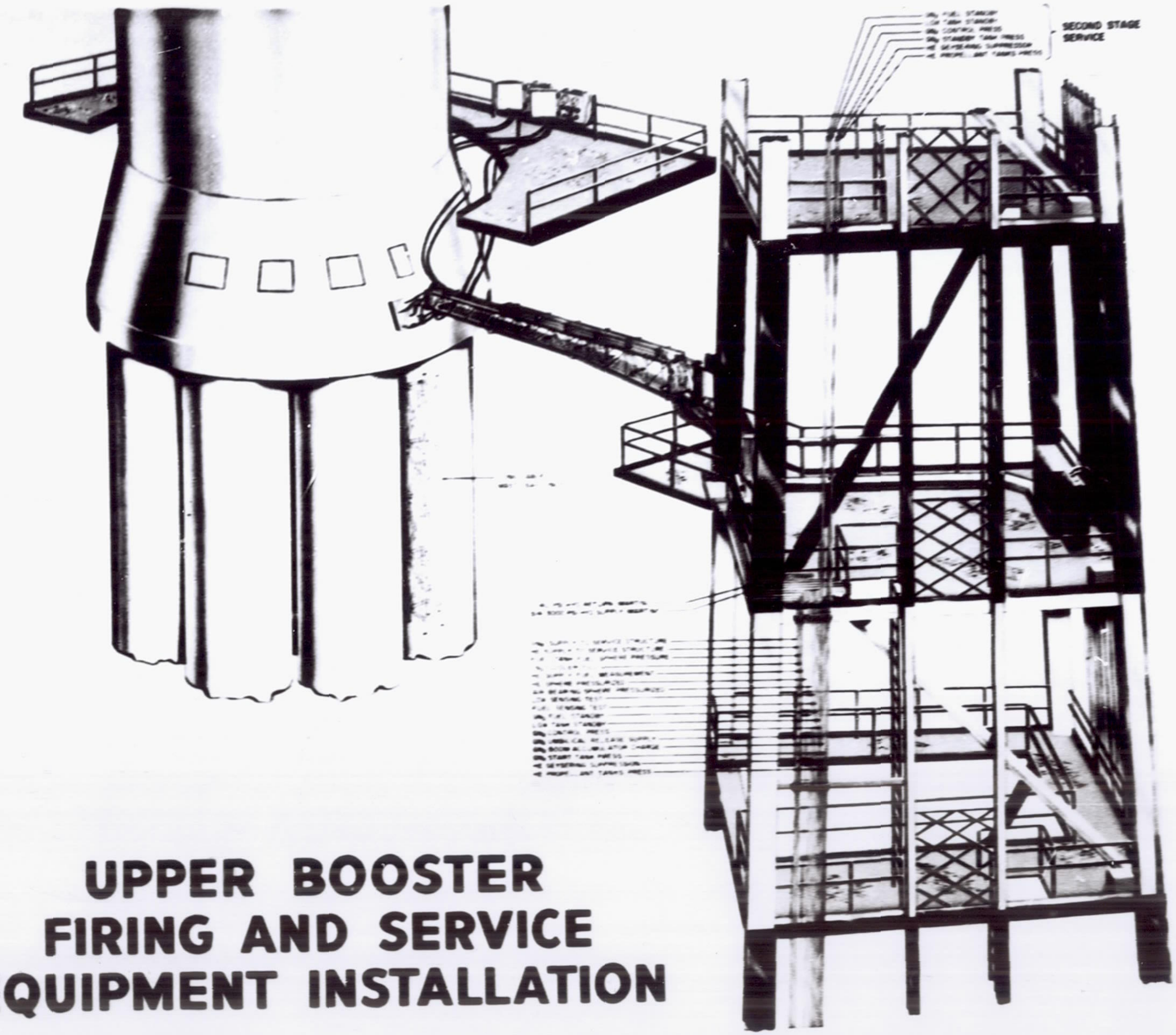


Figure 10.



UPPER BOOSTER FIRING AND SERVICE EQUIPMENT INSTALLATION

Figure 11.

Removal of an engine is accomplished by attaching a hoisting bar assembly to the engine "rabbit ears" and turbo pump mounting frame. The engine is then lowered by a hoist and pulley system until it is clear of adjacent engines and the booster structures. Here the engine is secured to a sled or skid which has been raised to position on a service platform. Half of the platform is then detached, lowered to the base of the launcher and removed from the area on a monorail. The engine is then lowered on its skid along the face of the deflector by means of a cable and hoist arrangement to the base of the launcher where it is removed from the area on the monorail. Reversal of this procedure is used to install a replacement engine.

5. <u>Upper Stage Assembly</u> (Figure 12)

Assembly of the upper stages is accomplished after booster checkout is completed. However, details on these operations are not yet available.

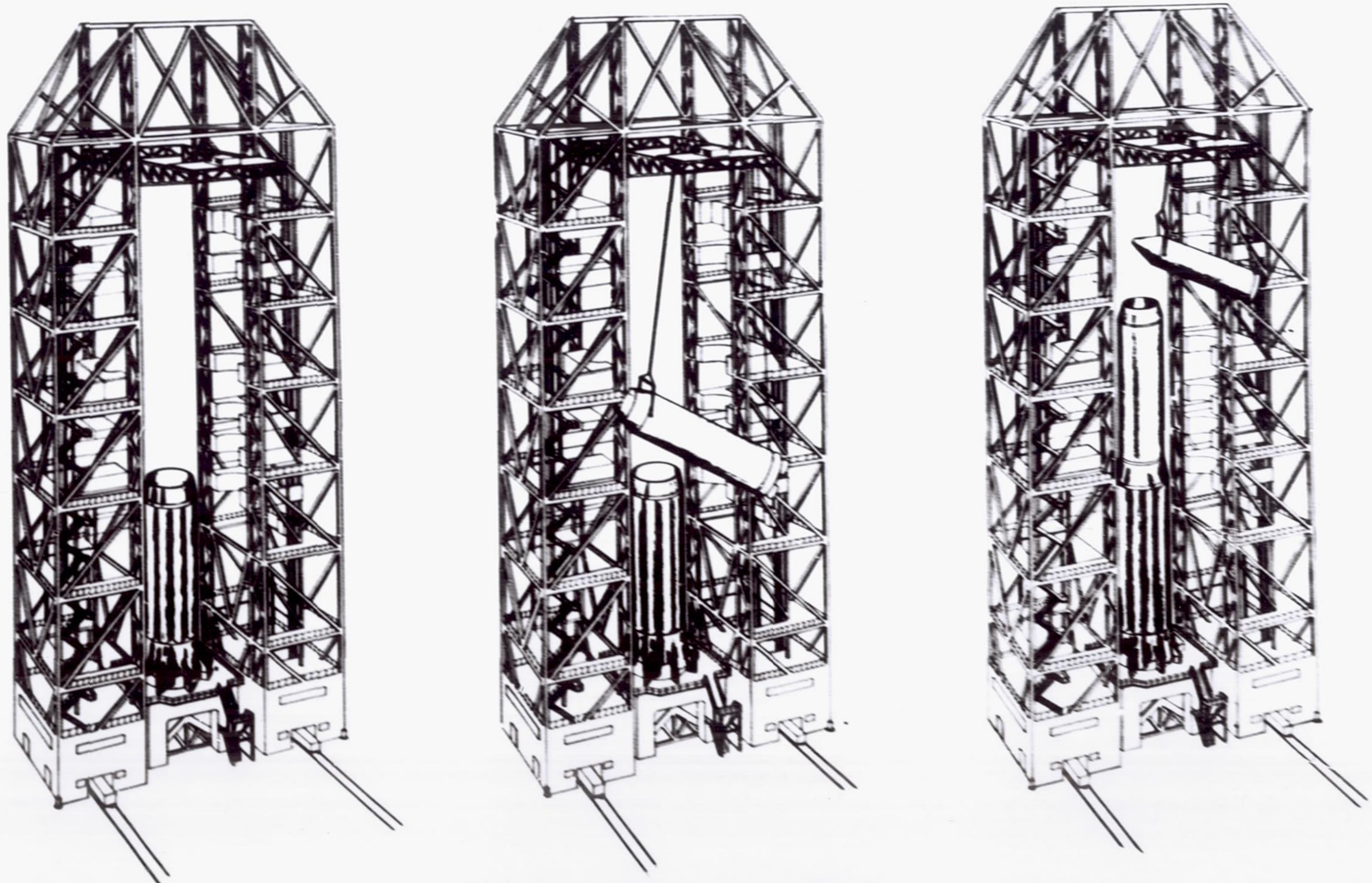
6. Launcher and Umbilical Tower

a. Launcher. (Figure 13) (Figure 14)

The SATURN Launcher is a reinforced concrete and steel structure 42 feet square and 27 feet high. It has eight support arms. Four supports 90° apart are cantilevered at the outboard engines and are retracted horizontally after the valid commitment signal is given to permit the engine shrouds to clear the missile during lift-off. The other four supports are dual purpose support and hold down points located at 45 degrees between the outboard engines. Hold down is accomplished by a toggle linkage which is activated when the retractable arms are all fully retracted. In event of malfunction of one or more of the retractable supports, all four supports may be returned to position under the missile thrust frame prior to engine cut-off.

b. Umbilical Tower. (Figure 15)

The umbilical tower is used to support and service the umbilical arms as well as to house and support the various electrical cables, pneumatic and LOX replenishing lines, liquid nitrogen cooling tanks, mechanical refrigeration units, the ground hydraulic unit, and the pneumatic and electrical distribution station which are required to service the booster and upper stages prior to launching. The tower is 240 feet high and 24 feet square at the base. The bottom 27 feet of the tower is enclosed to provide for two air conditioned equipment rooms. Above the 27 foot level the four-tower columns slope inward to a 10-foot square at the top. Tower facilities include safety ladders and service platforms at 20 foot intervals, a 2000-pound capacity personnel and small hardware elevator, and a 3000-pound capacity electric hoist at the top for handling lines, cables and the umbilical arms.



SATURN STAGE ASSEMBLY

Figure 12.

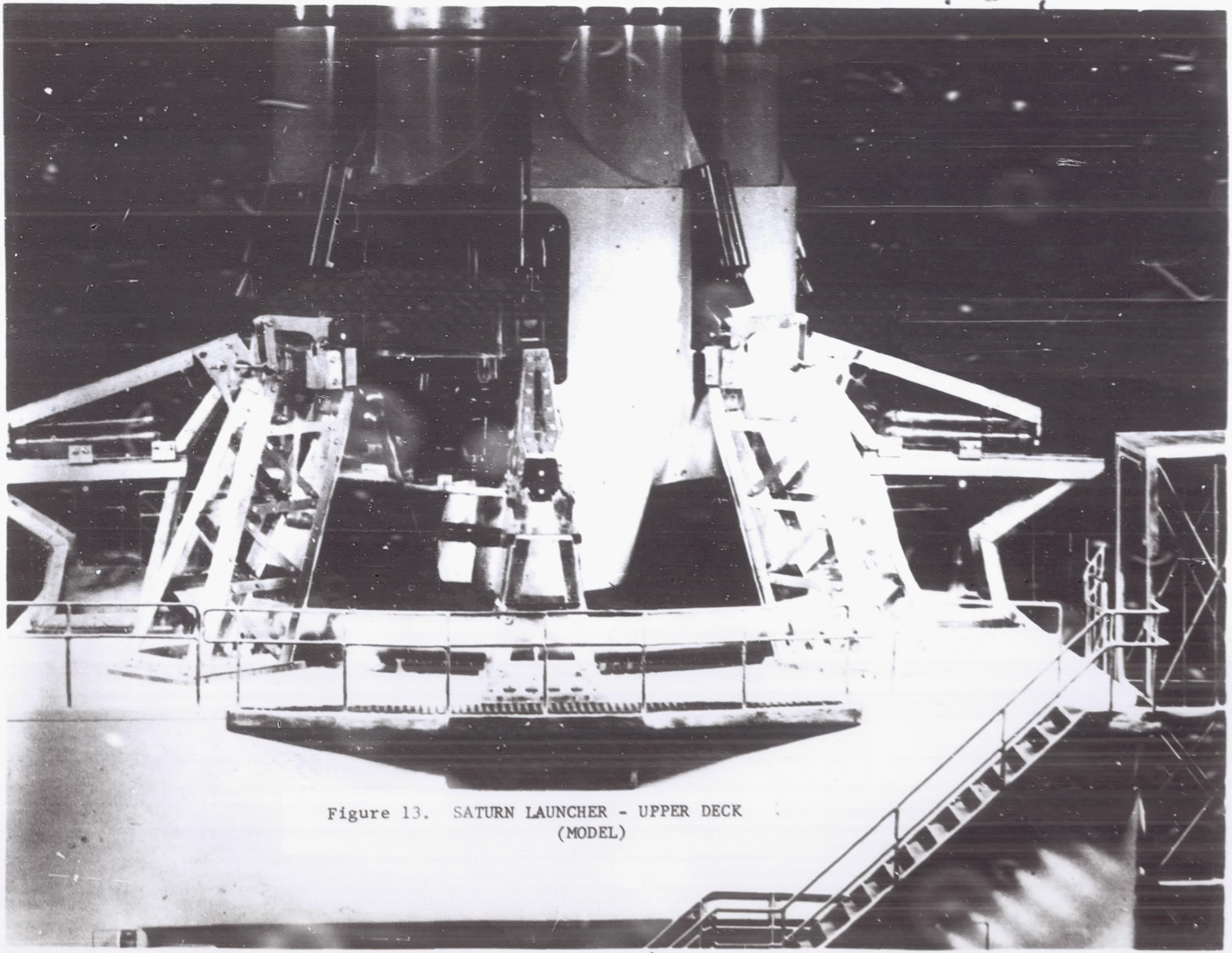


Figure 13. SATURN LAUNCHER - UPPER DECK
(MODEL)

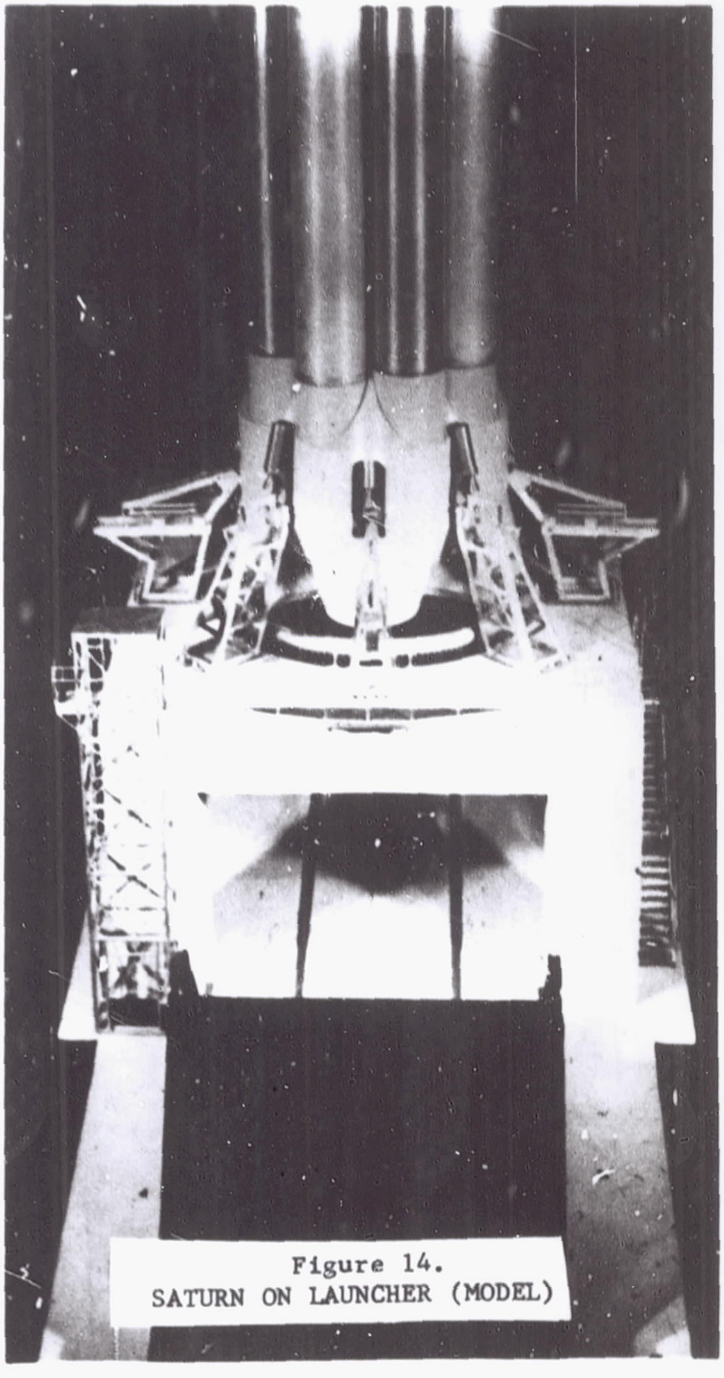
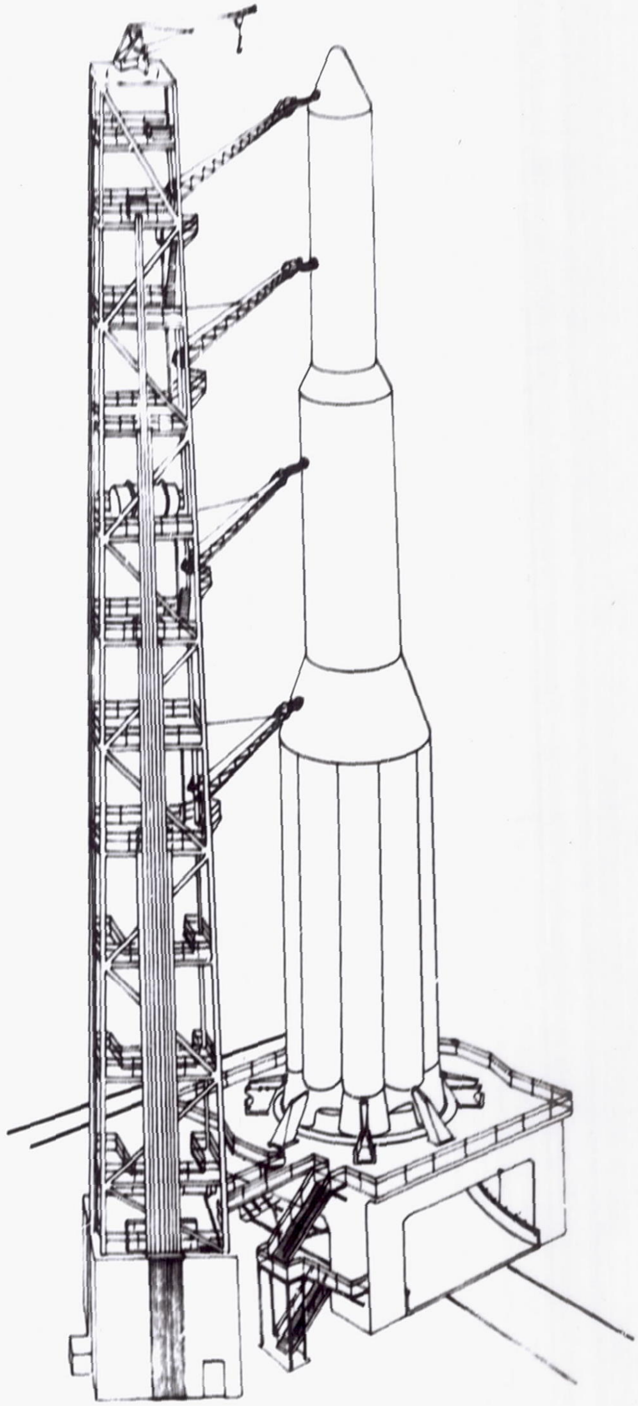


Figure 14.
SATURN ON LAUNCHER (MODEL)



THREE STAGE SATURN
WITH UMBILICAL TOWER

Figure 15.

7. <u>Propellant Storage and Loading</u>

a. <u>Liquid Oxygen</u>. (Figure 16) (Figure 17)

The LOX facility for the SATURN complex consists of protective revetments, foundation, and partial weather protection for liquid oxygen storage and transfer system. The system has the capability of servicing two pads from one facility at different intervals. The storage facility consists of an insulated sphere of 125,000 gallon capacity. The 41-foot diameter storage tank is insulated, but not vacuum jacketed, to sustain an evaporation loss of LOX of 0.2%/24 hr. The working pressure of the sphere is 40 psig which is maintained by a heat exchanger for self-pressurization. The booster transfer system consists basically of a 2500 gpm, 400-foot head pressure centrifugal cryogenic pump with associated valving necessary to transfer liquid through 750 feet of 8-inch uninsulated aluminum transfer line. Initial loading of oxidizer to the upper stages is facilitated by manifold lines connected to the gantry service structure and branching off at each stage servicing connection. The second stage initial filling is accomplished by using a 1000 gpm, 600-foot head pressure pump, and a 6-inch aluminum transfer line. The third stage filling utilizes the same 1000 gpm pump but is operated under throttled conditions.

Replenishing of the various stages is accomplished by using an additional 13,000 gallons, 200 psi working pressure, vacuum jacketed tank located in the storage facility confines. Replenishing of the booster and upper stages is accomplished by using a pneumatic actuated modulating valve controlled by a LOX tanking computer and level control associated with each of the stages to be replenished.

The replenishing transfer lines for the booster, second, and third stages are 3 inch, 2 inch and 1 inch insulated lines respectively.

Replenishing LOX to the upper stages is controlled to exact level and maintained until umbilical separation during the launch countdown. The 2 inch and 1 inch insulated replenishing transfer lines for the upper stages are routed through the umbilical tower.

The LOX transfer system is automated and is initiated and controlled from the blockhouse propellant loading panels during transfer sequence. Prior to propellant loading, the system component checkout can be accomplished at either the blockhouse or the LOX complex.

LOX STORAGE AREA-SATURN

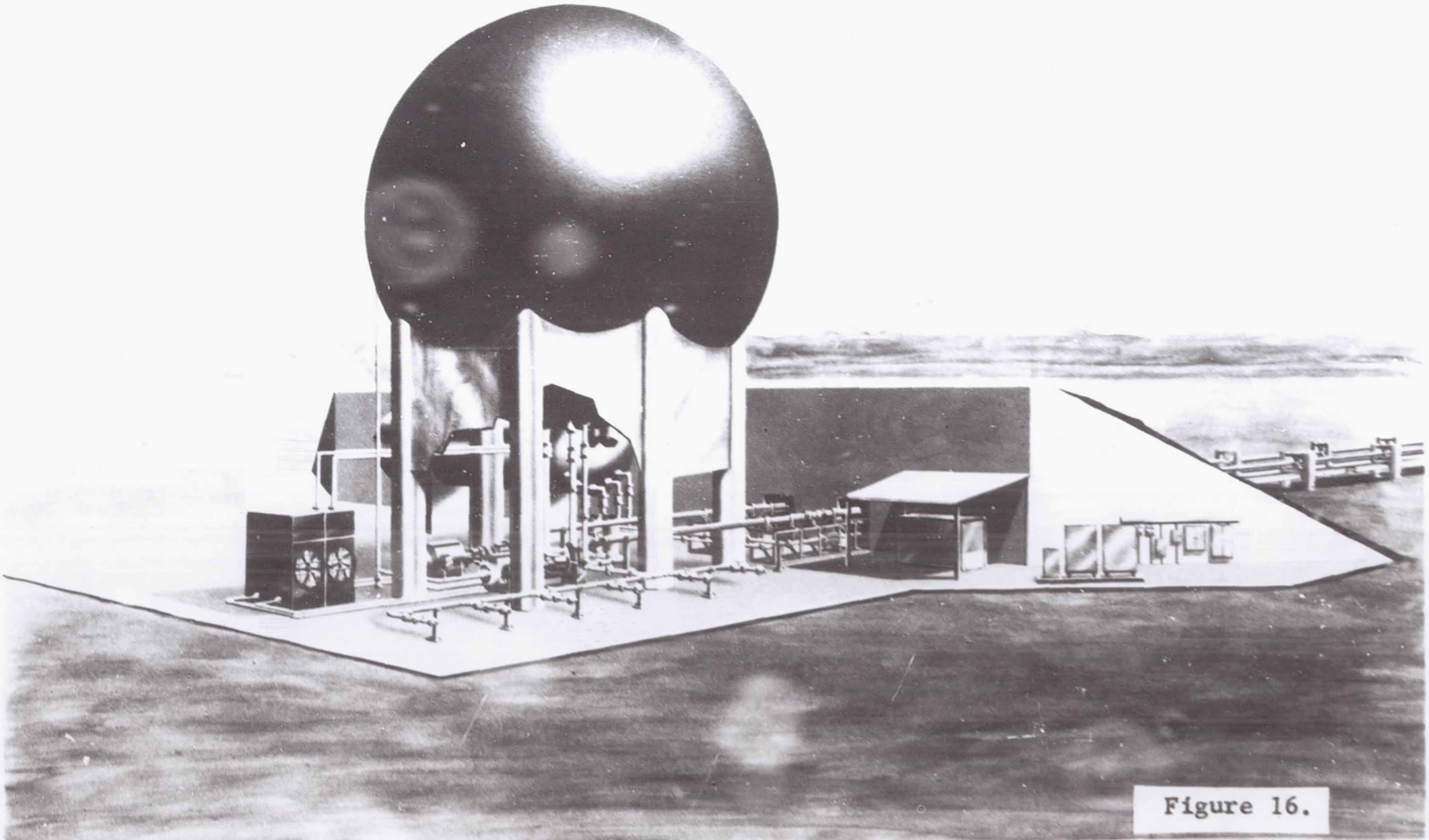
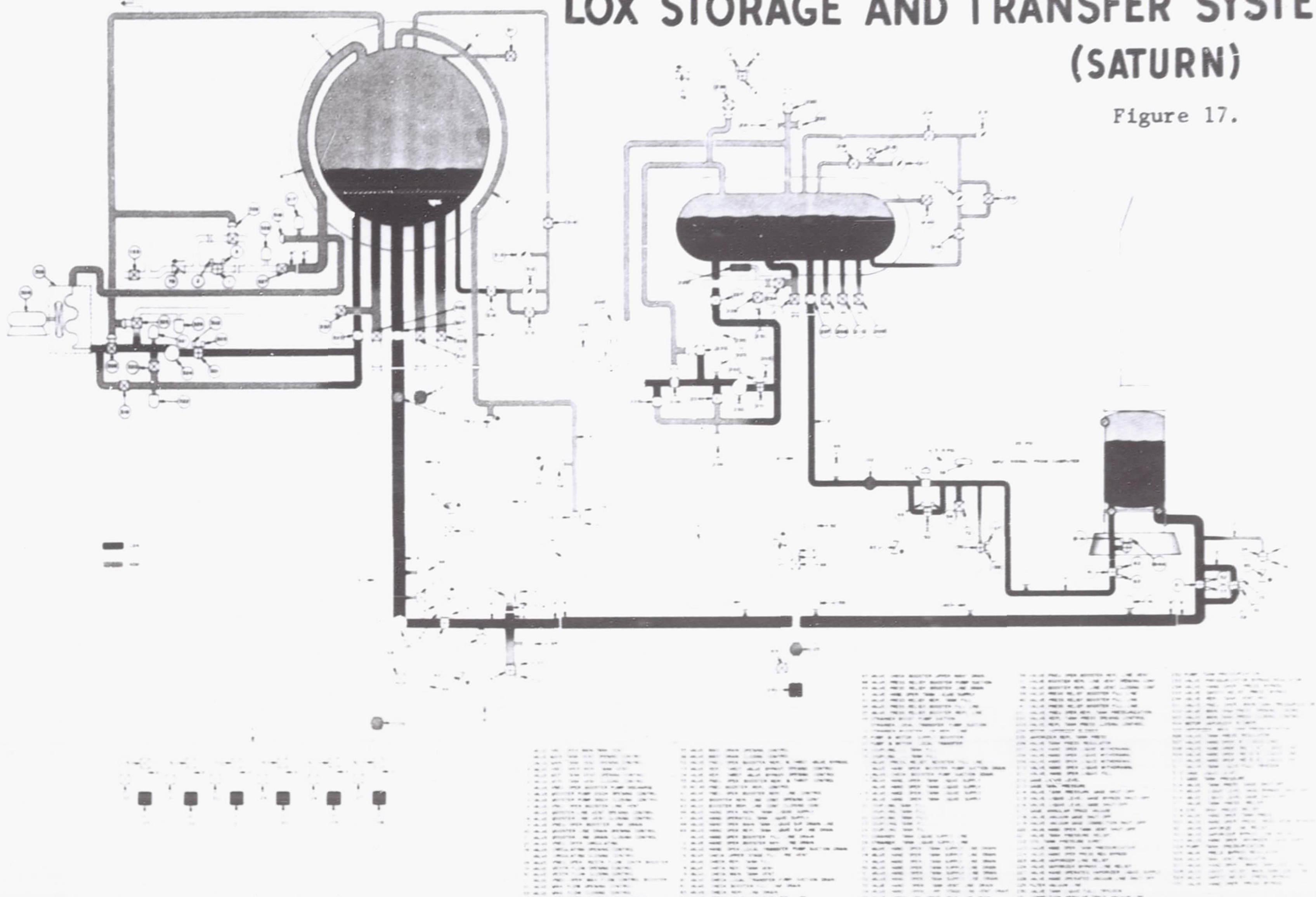


Figure 16.

LOX STORAGE AND TRANSFER SYSTEM (SATURN)

Figure 17.



b. <u>Fuel</u> (Figure 18) (Figure 19)

The RP-1 fuel facility for the SATURN consists of protective revetments, foundation, and partial weather protection for the storage and transfer system. The system can service two SATURN launch pads from one storage facility but at different intervals. A retaining wall to contain 125% of the volume of the fuel tanks is provided with the reveted area to retain the fuel in case of a tank rupture. The revetment wall is 15 feet high and is earth reveted on the pad side. Fuel storage is facilitated by using two 30,000-gallon capacity cylindrical insulated tanks. The transfer system and associated plumbing consist of a pad to support two 1000 gpm centrifugal pumps operating at 175 psi head pressure for fueling the booster, a 600 gpm recirculation pump, a 600 gpm filter-separator unit, an aductor system, miscellaneous valves, piping, and controls. The booster is serviced by two 1000 gpm pumps manifolded into 1000 feet of 8 inch diameter transfer line. Liquid level is controlled in the booster by a fuel tanking computer. The density of RP-1 fuel is monitored at all times by the fuel density indicator. The fuel is over-filled in order that the fuel computer can adjust fuel level to 100% by draining the fuel. The proper LOX-fuel weight ratio at take-off is accomplished by replenishing the LOX to the proper level as dictated by the fuel.

Initial upper stage fuel filling is accomplished through the same system but by utilizing only one 1000 gpm pump and a six-inch line attached to the gantry service structure. Once filled to the prescribed level the fuel transfer lines are evacuated of RP-1 by a jet eductor operating on Bernoulli's Law of Continuity.

Initial charging of the storage tanks is through the filter-separator unit to insure proper filteration of fuel and to minimize the entrained water content of the RP-1. During long storage periods, periodic operation of the filter-separator unit is required to insure desired fuel cleanness prior to missile servicing.

The fuel transfer system is automated and is initiated and controlled from the blockhouse fuel loading panels. Prior to propellant loading, the fuel system component checkout is accomplished at the blockhouse by the fuel loading panels. Communications between blockhouse and fuel complex is required during this operation.

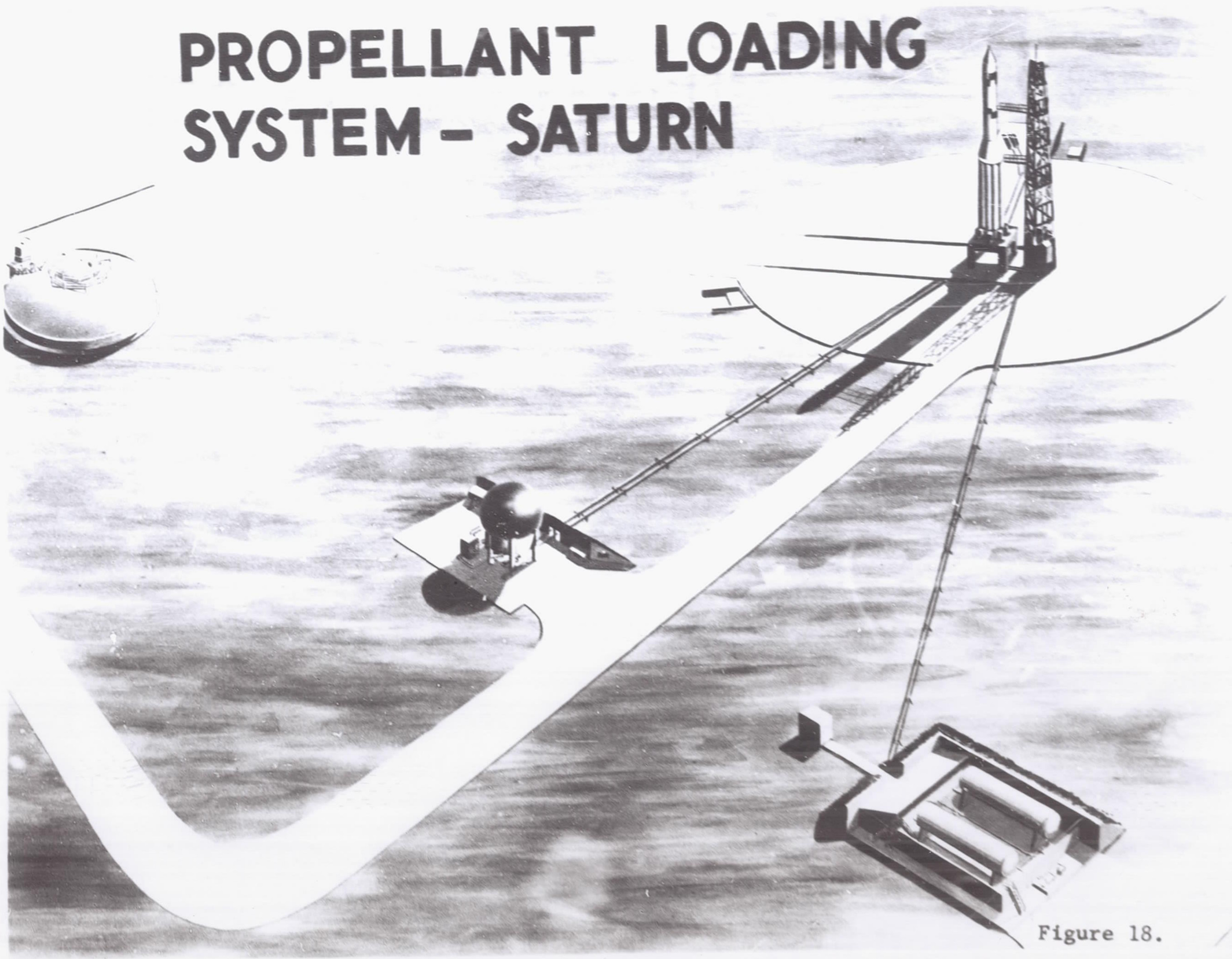


Figure 18.

FUEL SYSTEM SCHEMATIC
(SATURN)

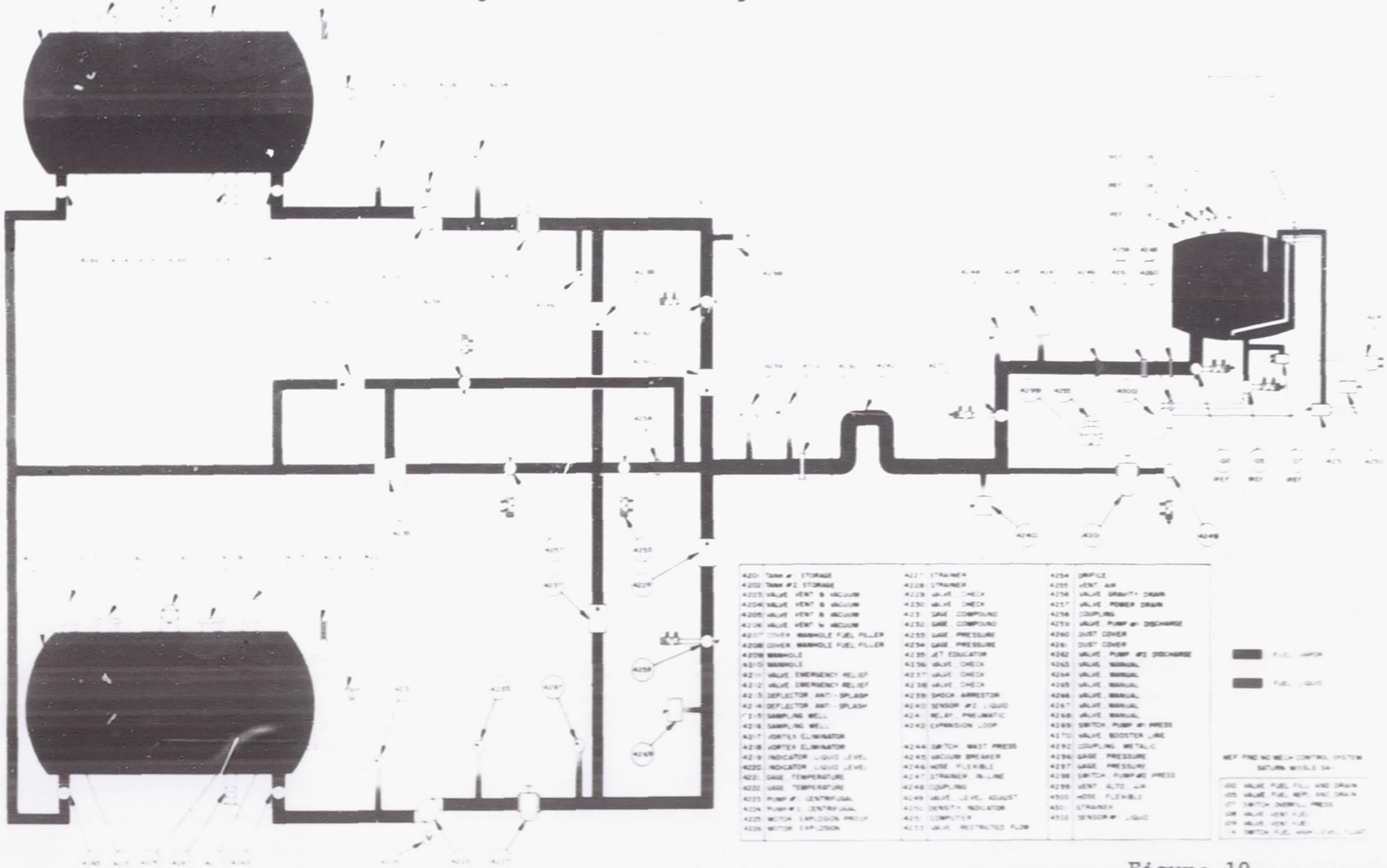


Figure 19.

35

D. BOOSTER RECOVERY OPERATIONS (Figure 20)

Recovery of the SATURN booster from the ocean is accomplished by the use of a fleet of surface vessels including an LSD (Landing Ship Dock), four destroyer (or similar) escorts, two sea tugs, a PT (Patrol) boat, and a command-communications-helicopter-tender ship of a suitable type. Fixed-wing aircraft are employed in spotting the downed booster.

The recovery operation will consist of four phases:

1. Location and damage surveillance
2. Recovery of the booster from the ocean
3. Decontamination and preservation and
4. Return shipment of the recovered components.

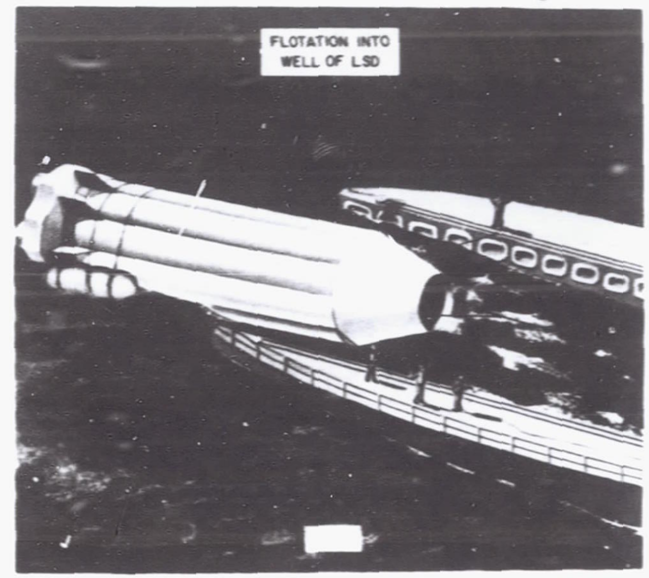
Immediately after impact, the helicopter and the PT boat seek out the booster and keep it under surveillance until the remainder of the recovery fleet arrives at the impact site. Upon arrival at the impact area, the recovery fleet is deployed for the recovery operation.

With the aft deck of the LSD awash, the booster is floated into position on fixed supports in the LSD well. The well is then pumped dry, leaving the booster in a supported position for decontamination and preservation.

Decontamination consists of a dry air or LN_2 purge of the LOX system, an over-all washdown with hot fresh water, and disassembly and cleaning of critical and special components. After decontamination, the booster and components are thoroughly dried by purging with hot, dry air and preserved by the application of desiccator breather assemblies.

The LSD then returns the booster to the appropriate dock (it is envisioned that this will be New Orleans) where it is barge-loaded for return shipment to RSA.

As stated previously, dependent upon the magnitude of damage to the booster during the recovery action, the booster would be put either on a transporter or on supports used in the LSD. The recovered booster or salvaged components, deemed reusable by inspection specialists soon after recovery, are transported from New Orleans to RSA by barge. There unloading is accomplished by roll-off of transporter, or when too heavily damaged, disassembly in barge and/or lifting the parts out of the barge by the available 20-ton crane.



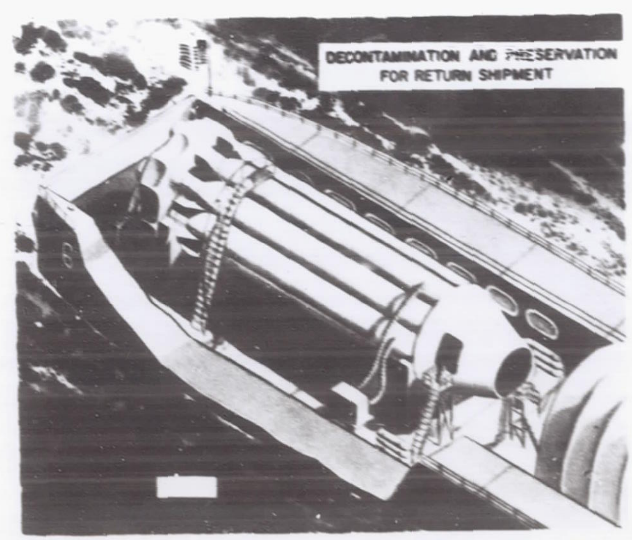


Figure 20. SATURN BOOSTER RECOVERY SEQUENCE

E. SYSTEMS SUPPORT EQUIPMENT LABORATORY TEST FACILITIES

1. Shaketable (Figure 21)

Test facilities available in Systems Support Equipment Laboratory include a shaketable for acceleration and vibration testing. The shaketable is capable of handling loads up to 10 feet in width and is adjustable to 50 feet in length. Combinations of three table sizes and adjustable shaft positions are possible. The table is a variable frequency (0-30 cps) fixed amplitude shaker and the amplitude is adjustable in progressive steps of one sixty-fourth of an inch up to one inch of single amplitude. The eight table bearings are capable of a total of 90,000 pounds of vibratory force. Therefore, items weighing 30,000 pounds can be tested to vibratory accelerations of 3g's.

Table frequency is monitored by a shaft tachometer and/or gear ratio and engine tachometer. Control is obtained by use of a hand throttle on the 183 horsepower industrial reciprocating gasoline engine. Automatic frequency control is available for special requirements. Auxiliary electronic equipment is available for measuring and recording acceleration, velocity stress, amplitude and frequence during shaketable operation.

Competent engineering and technical personnel are available to design, set up, instrument and evaluate tests as required.

2. Propellant Storage and Transfer

Facilities presently available in Systems Support Equipment Laboratory for propellant handling, storage and transfer include the following major items of equipment:

a. Loading and Measuring Test Tower. This is an 80' tower supported on a 100'x 150' concrete pad with a 160,000 pound capacity platform scale for use in loading and measuring tests.

b. Blockhouse. A reinforced concrete earth-covered structure houses personnel and measuring, monitoring, recording and control equipment to support the test tower operations. Operations at the LOX and fuel facilities may also be remotely controlled from this blockhouse.

c. LOX Transfer Facility. A 60' x 60' concrete pad, one-half under roof and one-half in open, supports one 4000 gallon and one 6000 gallon capacity stainless steel, insulated, 100 psi working pressure LOX storage tanks for use in both pressure and pump transfer tests. Also on this pad is one 6000 gallon capacity, insulated, 200 psi working pressure LOX storage tank which is used in LOX replenishing testing as well as pump and pressure transfer tests. The pad has a 3600 psi, -65°F dew point air supply and electrical power supply to operate motor-driven pumps up to 350 horsepower.

d. LOX Storage Facility. One 75' x 150' concrete pad with four 35 ton vacuum jacketed LOX storage tanks for low loss storage. The pad is adjacent to railroad spur track on which LOX supply is delivered by rail tankcars.

e. Fuel Storage and Transfer Facility. One 60' x 60' covered concrete pad with one 7000 gallon and two 10,000 gallon RP-1 storage

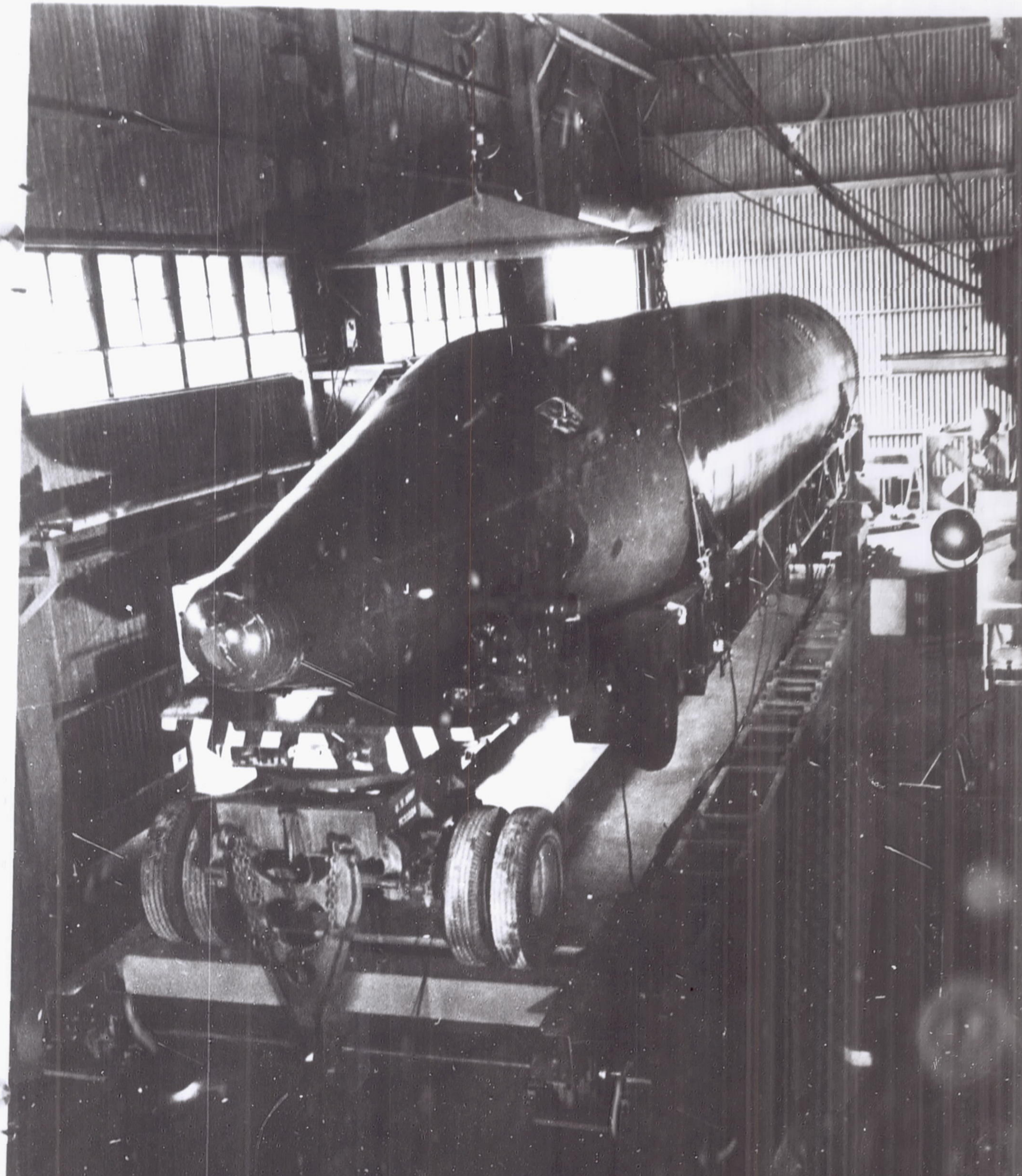


Figure 21. SHAKETABLE

containers. The pad also has a fuel filtering and dewatering unit with a capacity of 200 gallons per minute, a power supply to operate motor driven pumps up to 350 horsepower and a 3600 psi, -65°F dew point air supply.

f. Transporters. Facilities for local propellant transportation and test include:

(1) Three 6000 gallon RP-1 transporters w/1000 gpm pumps at 60 psi head and heaters for temperature conditioning of fuel.

(2) One 3000 gallon alcohol trailer w/250 gpm pump.

(3) One 5000 gallon kerosene trailer w/250 gpm pump.

(4) Two 7½ ton LOX transporters with pumps having 150 gpm at 75 psi capacity.

(5) Two 9 ton LOX transporters with pumps having 150 gpm at 75 psi head capacity.

(6) Six 4000 gallon LOX transporters.

(7) Two 500 gallon liquid nitrogen trailers with high pressure pump that deliver 4800 scfm of gaseous nitrogen at 5000 psi.

g. Transfer Equipment. Additional miscellaneous transfer equipment available includes:

(1) Two motor-driven transfer pumps with 1500 gpm at 100 psi head capacity.

(2) Transfer piping up to 8" diameter and 800' length for transfer tests at capacities up to 2500 gpm.

(3) Two remotely operated LOX transfer trailers with electric motor-driven pumps having a capacity of 1500 gpm at 100 psi head. The trailers are equipped with LOX tanking computers and system controls

for accurate missile loading ($\pm$ 0.1%).

(4) Six fuel volumetric flow meters ranging from 250 - 600 gpm capacity with an accuracy of $\pm$ 0.1%.

(5) One electric motor-driven fuel pump with a capacity of 600 gpm at 60 psi head.

(6) One missile simulator receiver with a 12,000 gallon capacity for use in propellant transfer and precision loading tests.

(7) One portable filter-separator with 200 gpm capacity.

h. Additional equipment and instrumentation necessary for temperature, pressure, flow, power, measurements and recording of data are available within the laboratory.

i. Competent personnel with broad experience in the field of liquid propellant handling, storage, transfer and transportation are available within the laboratory.

Design criteria for a new propellant testing facility for Systems Support Equipment Laboratory have been proposed. The new facility includes the needs and requirements of the laboratory for projected future test program for the SATURN missile. The new facilities would be constructed on a new site apart from existing facilities and would include a propellant test tower; all propellant storage and transfer requirements, complete blockhouse propellant testing and recording equipment, power supply and transportation access.

F. GROUND TEST PROGRAM

1. Propellant Loading System

Many new items will be required for the SATURN propellant loading system which must deliver over eight times the amount of propellants required by the JUPITER, and with comparable accuracy. These items will be tested individually before incorporation into the propellant system.

2. Transporters

a. Booster Transporter (Figure 22). A booster transporter, carrying a dummy booster, will be subjected to several road performance tests to assure satisfactory braking, steering, turning, and general handling characterists before it is used for the first road movement of a flyable booster.

b. Payload Transporter (Figure 23). A complete test program will be carried out on the payload transporter and handling equipment to determine the adequacy of those items requiring special design. A dummy payload will be used in road tests and in simulated aircraft loading tests to assure satisfactory steering, turning, and handling characteristics.

3. Dummy SATURN Booster

a. The dummy booster was fabricated by modifying salvaged steel tanks and welding them together to simulate the SATURN booster weight, center of gravity, and overall length. The spider section with attached fabrication and assembly rings are required to give the simulated cross sectional diameter to the dummy as well as to serve as erection rings and support of the missile when in the transporter saddles.

Weight at pick up and support points is simulated by filling water tanks over these points to the desired weight. The test equipment mounted on the transporter will be loaded and off-loaded from the barge to familiarize personnel with the transfer sequence prior to the actual transfer of the flyable booster. The dummy booster will also be utilized to check out the fabrication and assembly jig as to rotational drive capabilities due to eccentric load applied by the 70" tanks during assembly. Road test to verify steering, braking, towing, and road conditions prior to shipment will be checked with the dummy booster on the transporter.

b. The dummy booster will also familiarize static test tower personnel with the problems associated with erection, stabilization, and operation of the 100 ton crane located at the ABMA static test facilities prior to receipt of the live static test booster.

4. <u>Launcher</u>

As parts vital to a successful launch operation, the launcher support and hold-down arms will be subjected to structural and functional tests at ABMA prior to assembling them on the launch pedestal.

5. <u>Handling Equipment</u>

Individual items of booster upper stages and payload handling equipment will be tested prior to and during environmental tests and actual erections and assembly at both ABMA and CCMTA.

6. <u>Booster Checkout Equipment</u>

All components of the checkout equipment will be laboratory tested and service tested during reliability checkout and static test firings at ABMA before use at Cape Canaveral on a flyable booster.

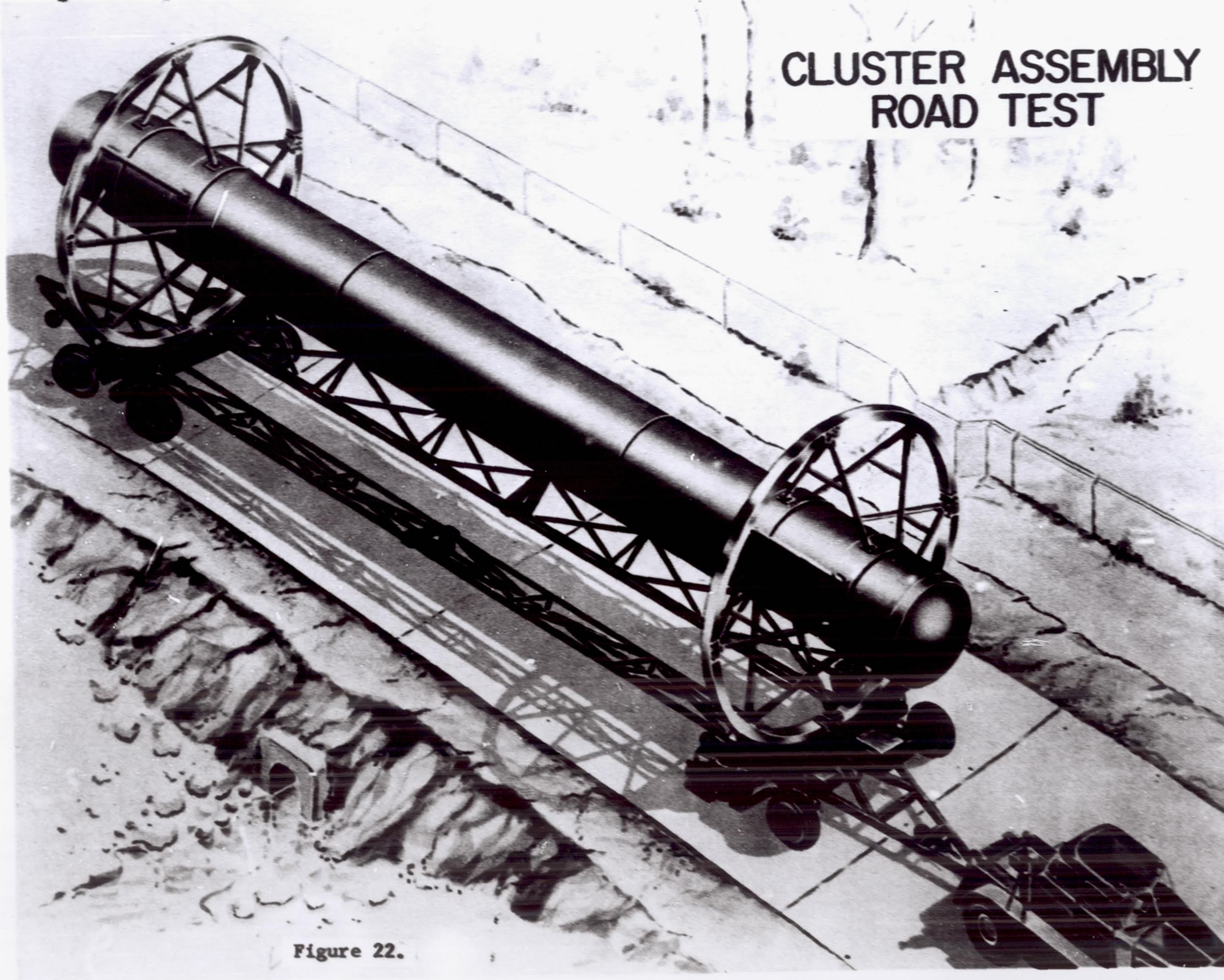
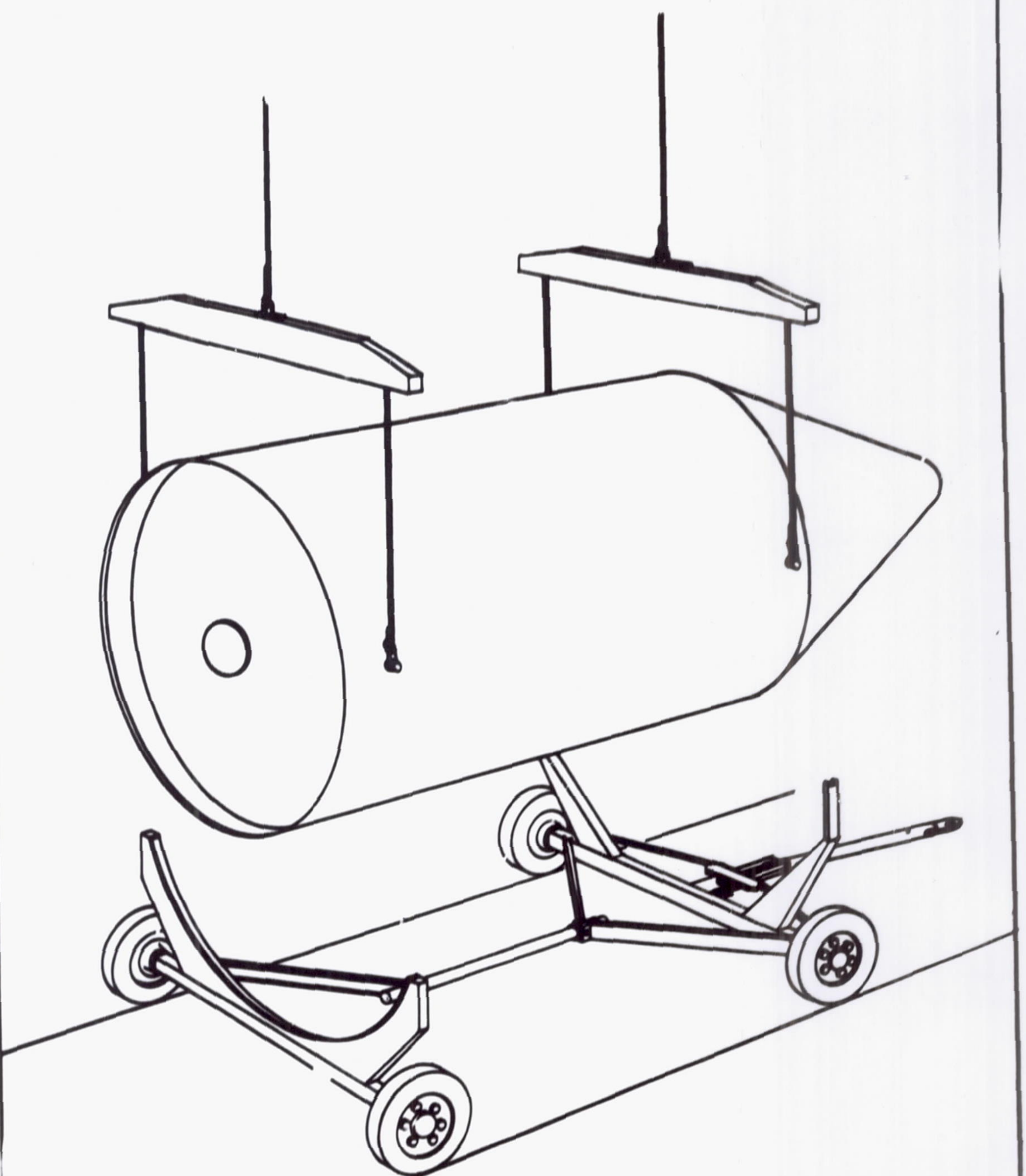


Figure 22.



ORIGINAL DATE OF DRAWING 4 SEPT 59	HANDLING OF	ORDNANCE CORPS	
DRAFTSMAN	CHECKER	DEPT OF THE ARMY	
TRACER	CHECKER	ARMY BALLISTIC MISSILE AGENCY	
ENGR	ENGR		
SUBMITTED	PAYLOAD		
ORD CORPS	Figure 23.		
APPROVED BY ORDER OF THE CHIEF OF ORDNANCE			
ORD CORPS			

7. Vertical Engine Removal Equipment

This equipment will be tested in the laboratory and at the static test tower prior to shipment to Cape Canaveral.

8. Booster Recovery

In order to study that phase of booster recovery in which the booster approaches and enters the water, the contractor for the SATURN recovery system has designed a scale model of the booster which simulates the center of gravity and pitch moment of inertia. This model will be dropped, with various velocities and angles of entry, into water having wave motions. These tests will determine the booster tipover and floating characteristics as well as the inertial loading at various booster points during impact. After these minimum tests are completed, parachute drop tests from different altitudes will be conducted, using about one-fourth of the SATURN booster mass and one chute. These drop tests will determine the opening and reefing characteristics of the large parachute and the shock loads experienced by the simulated booster during these events.

Other recovery package components such as timers, pyrotechnic devices, aneroid switches and motors will be tested under environmental conditions to assure satisfactory performance of the recovery system under operational conditions.

LEFT BLANK INTENTIONALLY

SECTION II

MANNED LUNAR CAPSULE RECOVERY
(Water Phase)

A. INTRODUCTION

1. General

A speedy, reliable and efficient system for recovery of the manned lunar capsule from the ocean after return from a lunar circumnavigation flight is required. The utmost speed and reliability are absolutely essential in this operation. The "loss of the horse for lack of a shoe" would be intolerable at this point. The proposed equipment and techniques are based on experience gained through use of present "state of the art" equipment in the actual recovery of the Jupiter and Jupiter "C" nose cones. The equipment presently aboard typical navy rescue and/or fleet tug vessels was originally designed for salvage of large stranded vessels and/or submarines and is not of optimum design for retrieving comparatively small nose cones or capsules. The proposed system of recovery offers what is believed to be the most infallable method of capsule recovery over the widest range of sea conditions and capsule configurations. The system may be readily optimized for any given capsule configuration from 120 to 160 inches in diameter, from 16 to 25 feet in length, and from three to six tons in weight. This system utilizes a hydraulically controlled crane together with specially designed retrieving and handling devices.

2. General Engineering Desiderata

Based on past experience in nose cone recovery and an evaluation of the advantages and disadvantages of the present equipment and techniques

the following general characteristics and requirements for a manned lunar capsule recovery system may be stated.

a. Fail-safe method of tracking and spotting the descending capsule.

b. Rapid transport of retrieving equipment to the point of impact.

c. Speedy deployment of retrieving equipment.

d. Versatile equipment for operation under extreme variations in sea conditions.

e. Fail-safe method of securing the floating capsule.

f. Sensitive and rapidly reacting equipment controls with the utmost reliability.

g. Short coupling between recovered capsule and retrieving equipment.

h. Speed and simplicity in equipment operation.

B. SYSTEM DESCRIPTION

1. <u>Recovery Fleet</u>

In the present state of the art the precise point of impact of a returning capsule cannot be controlled. Therefore, extensive fleet deployment in the general impact area may be necessary. The deployed fleet consisting of recovery and auxiliary ships must be equipped with both radar and visual detection equipment and be augmented by orbiting aircraft. Predictable impact accuracy will determine the number of ships in the fleet that must be equipped with recovery equipment.

2. <u>Recovery Ship</u> (Figure 24 and 25)

The actual recovery ships must be fast and highly maneuverable.

Ships of the new destroyer class seem to best meet these requirements in addition to having sufficient sea worthiness to navigate and operate in any sea condition in which a recovery could conceivably be attempted. This class of ships and the general arrangement of recovery equipment on board is shown in Figures 24 and 25.

3. Underwater Teams (Figure 25)

Each recovery ship should be provided with two teams of frogmen equipped to perform underwater inspection and to attach special retrieving gear when necessary. The equipment for each of these teams should include a motorized rubber life raft and reliable 2-way radio communication with the mother ship. The diving team must be trained to remove the parachutes, marker buoys and any other impedimenta which might interfere with recovery operations. During actual retrieving operations under normal conditions the frogmen are not required to perform any tasks involving physical contact with the capsule or equipment. This is an important safety consideration, especially when operations must be performed in rough sea conditions.

4. Hydraulic Crane (Figure 25)

The proposed lifting device for removal of the capsule from the water and placing it on the ship's deck is a hydraulic crane of the Bucyrus-Erie type as shown in Figure 25. The hydro-crane is in effect a stiff arm that may be moved in any horizontal or vertical direction. The hydro-crane mechanisms are actuated by high pressure hydraulic fluids ranging up to 6000 psi working pressures. All crane functions, including boom swing, lowering, retracting and hoisting are controlled by simple levers and any operator who is familiar with general

crane operation may be taught to operate this crane in a very short time. Attachment of this type crane to the ship's deck is accomplished simply by the attachment of the four deck plates and it is felt that shifting of this crane from one ship to another would be entirely feasible if required. The crane boom must be modified to provide a jib boom approximately seven feet in length as shown in Figure 25. When equipped with two hoisting lines, this modification will permit the application of present securing devices and will be versatile enough for other future applications. The hydro-crane fulfills the stated requirements for speed and simplicity of operation, reliability, versatility, short coupling capability, and sensitive precision controls.

5. Hoop Net (Figure 25 and 26)

A hoop net is proposed as the retrieving device to be used in conjuction with the hydro-crane. The hoop net is cylindrical in shape and should have a diameter and length approximately four feet greater than the capsule to be recovered. Determination of final dimensions will depend upon the capsule dimensions and whether or not it will float on its side or nose. Dimensions of the nets may be varied for recovery of different sized capsules but operating characteristics will remain the same. The net may be constructed of any fibrous material, such as hemp or nylon, with sufficient tensil strength to carry the load of the particular capsule to be recovered. The net is held in shape and given body by incorporation of several collapsible elastic hoops along its length. When extended these hoops hold the net in shape while it is lowered over the capsule. Weights or sinkers placed near the bottom of the net cause it to sink rapidly over the capsule.

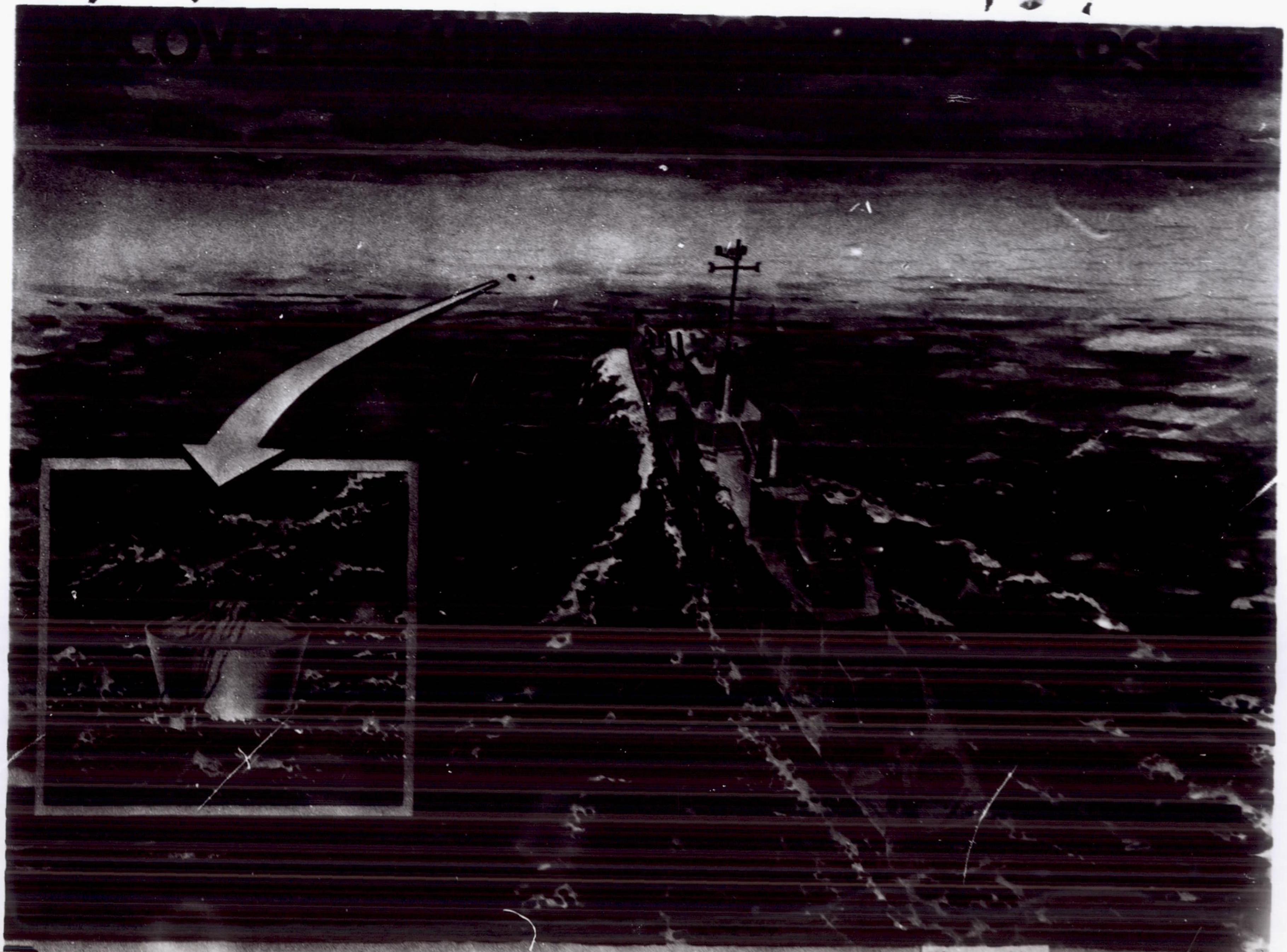


Figure 24.

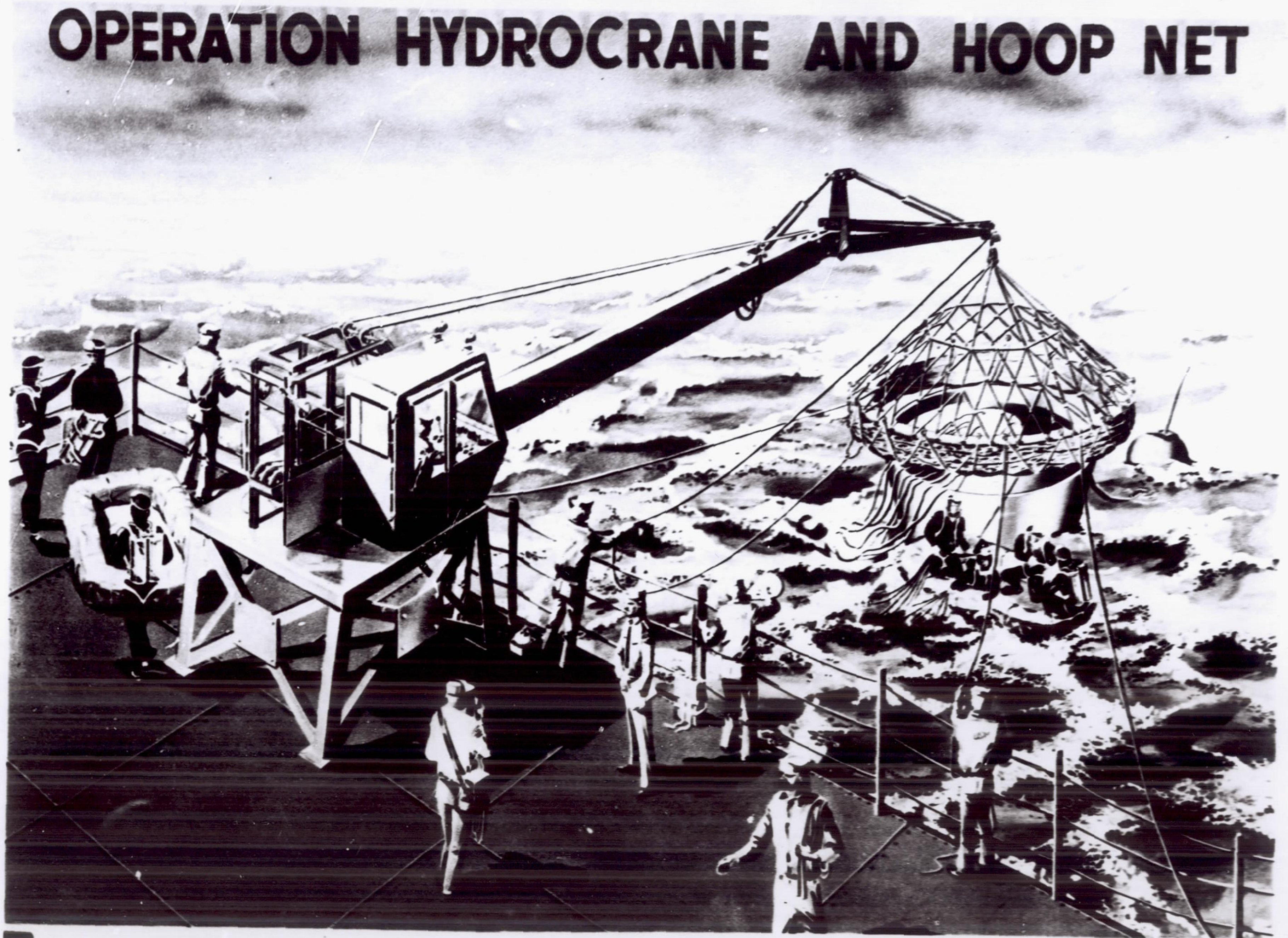


Figure 25.

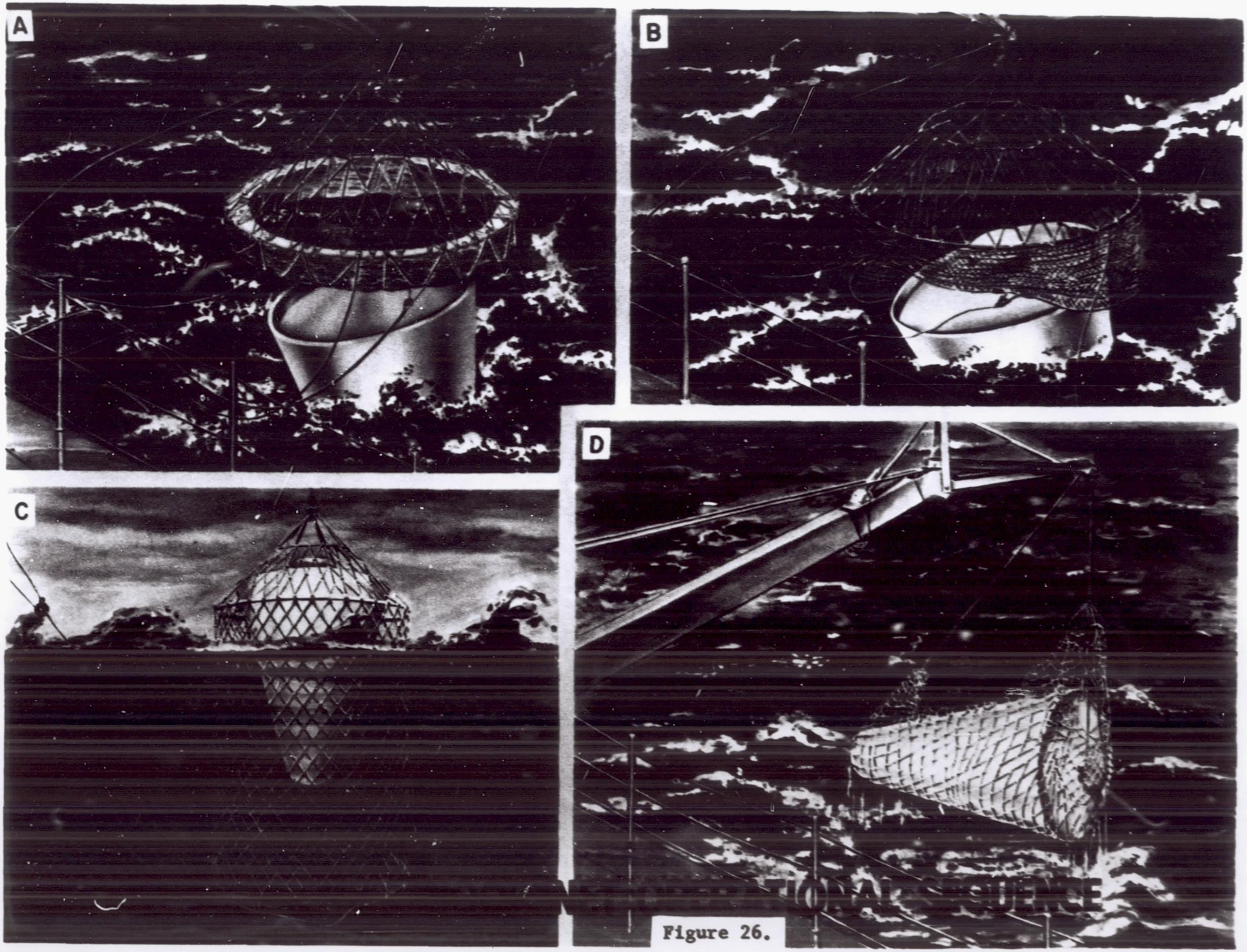


Figure 26.

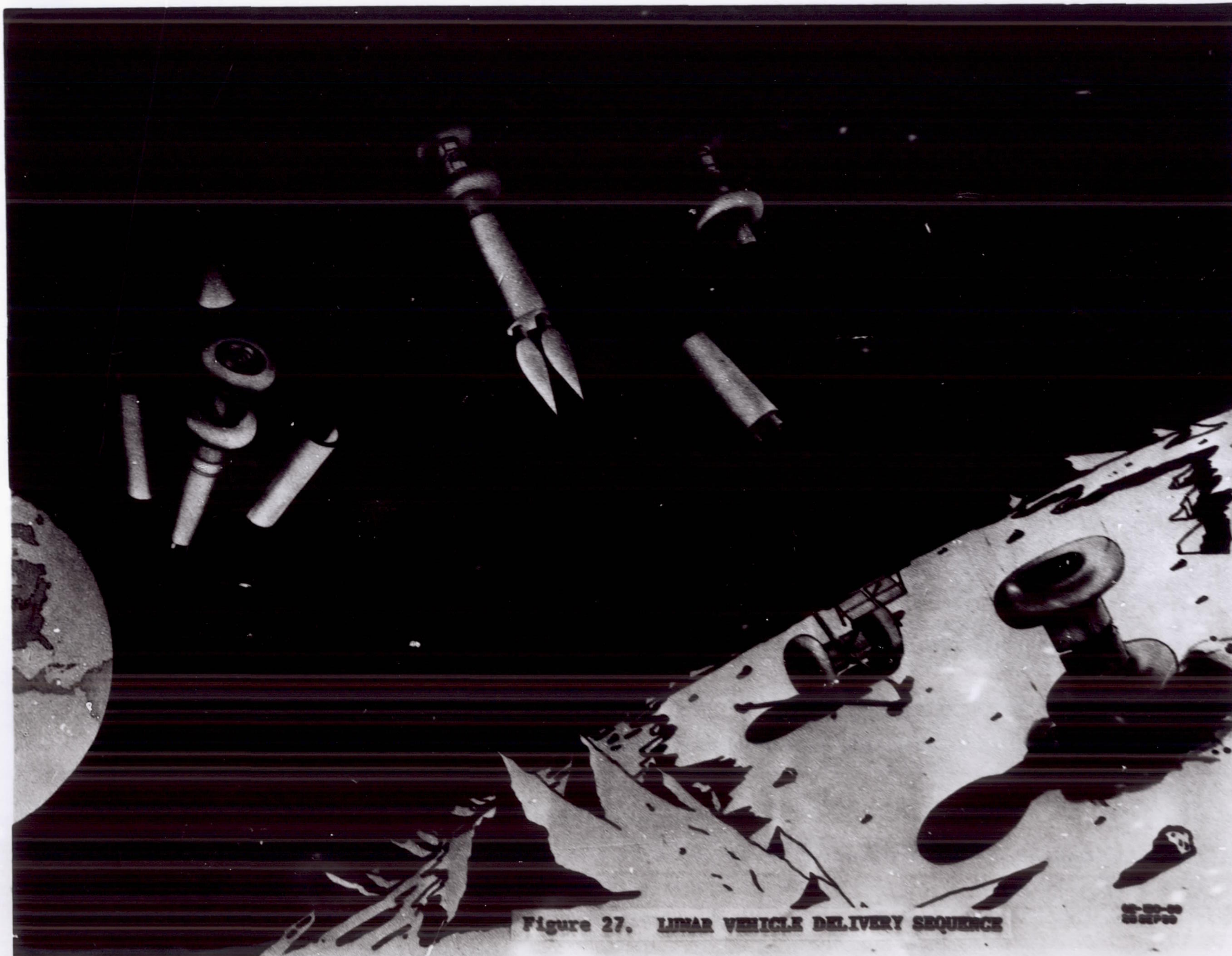


Figure 27. LUNAR VEHICLE DELIVERY SEQUENCE

The bottom or open end of the net will be equipped with several cable clinches with undirectional action through which a gathering or closure cable will be operated to effect closure of the net. The closure cable will be attached to a boom line which may be temporarily secured aboard ship during the lowering operation. After the net has been lowered over the capsule, a weight of about 100 pounds attached to a pulley is placed on the boom line and allowed to run down the cable and effect closure of the net by its inertia. The weight is then pulled back up the cable by a retrieving hand line and detached from the cable. With the capsule thus secured in the net the crane operator may remove all slack from the line connected to the closure cable and lower the boom to reduce the distance between boom tip and cargo to effect a short coupling which, together with jib boom design, will minimize the pendulum and gyrating effects of the capsule and net. The capsule and hoop het are then hoisted aboard the ship with the capsule in approximately a horizontal position. This operation is illustrated in Figures 25 and 26. The hoop net fulfills the stated requirements for speed of deployment and operation, versatility, reliability, and fail-safe method of securing the capsule. In addition, it has operational capabilities over a wide range of sea and weather conditions as well as capsule conditions such as distortion and severe structural damage.

6. Operational Sequence

Normal operational sequence for the proposed recovery system is outlined as follows:

a. Location of capsule.

b. Nearest recovery ship proceeds to point of impact (Figure 24).

c. Deployment of frogman team (Figure 25).

d. Removal of parachutes and floatation buoy.

e. Positioning of recovery ship and hoop net (Figure 26A).

f. Release of net (Figure 26B).

g. Closure of net (Figure 26C).

h. Retrieval of capsule (Figure 26D).

C. PROJECT IMPLEMENTATION

It is proposed that the R&D effort for this phase of the program be initiated in sufficient time to permit the production and preliminary test of at least one complete set of recovery equipment as described above prior to the capsule drop test and training program. It is further proposed that when the capsule drop tests are conducted the water recovery phase be incorporated in this test and training program. This will permit a realistic test and provide training on the entire recovery operation techniques and equipment well in advance of a live recovery. In this way much valuable experience and training will be gained at minimum additional expenditure of funds and effort and required modifications can be accomplished early in the program.

SECTION III

PROPOSED LUNAR ROVING VEHICLE

A. INTRODUCTION

1. General

Two types of soft landing payloads are proposed for initial lunar surface exploration. The first soft landed payloads will be stationary packages designed to gather data at the point of impact. These packages will be followed later by a soft landed vehicle designed to rove a restricted area of the lunar surface in the vicinity of its point of impact and gather data on surrounding conditions. To some extent, the stationary payload to be landed will serve as an R&D vehicle for improving design of the roving vehicle components. For example, it is proposed that the crush bag or deceleration equipment should be the same for both payloads. Other components and characteristics of the two payloads will also be the same or similar. However, this section is devoted primarily to a description and discussion of the proposed roving vehicle. Material presented is not intended as final data, but is the results of considerable preliminary investigation of the anticipated problems and will serve as the basis for the detailed R&D program that will be necessary before final design can be accomplished. This proposal is made with full awareness of the vast amount of research and development work presently being performed by many agencies in the field of small power plants and the many investigations of new applications. However, it is felt that at the present state of the art the proposed system affords the best solution to the problem. This section also contains recommendations for the vacuum test

facilities necessary for the R&D program.

2. General Engineering Desiderata

The general characteristics and requirements listed below have been established for the lunar roving vehicle and have been considered in the proposed design.

a. Withstand impact on the lunar surface with a velocity of approximately 60 ft/sec with maximum deceleration of 20g (earth).

b. Operate under lunar environmental conditions as a mobile vehicle for one full lunar day (daylight).

c. Have a range of 50 miles, be able to negotiate a slope of 15°, pass over boulders 3 - 4 feet in diameter, and maneuver to avoid larger boulders or hazardous terrain.

d. Due to uncertain surface conditions, it should be able to travel on thick layers of dust.

e. Total payload package weight limit - 2175 pounds.

f. Total roving vehicle weight limit - 1450 pounds.

3. Lunar Vehicle Delivery Sequence (Figure 27)

During the first stages of the missile flight the lunar vehicle will be housed within the payload section of the missile third stage which will have an outer diameter of 10 feet. Therefore, the tires and certain operating equipment must be in a collapsed condition. Prior to the beginning of the lunar landing phase of the payload trajectory, the nose cone and protective cover (airframe) will be blown off the payload package by explosive charges triggered by a timing mechanism. This action will be delayed as long as feasible to afford maximum meteorite protection to the vehicle. When the covers are blown off the

vehicle tires will be free to expand immediately to their full size at operating pressure. The vehicle will remain in this attitude throughout the braking and landing phases. Attitude control will be provided during these phases to insure that the payload package lands in an upright position and with minimum lateral motion. When the landing phase is complete and the vehicle has impacted on the lunar surface, it will topple over from the vertical axle attitude to a horizontal axle traveling position. The off center CG of the vehicle instrumentation package will serve to initiate this toppling motion. Blow off or spring loaded devices used to separate the shock absorber and final guidance and control equipment from the vehicle may be used to assist in initiating this toppling action. Once the vehicle is in normal horizontal traveling attitude the torque reaction arm, the turbo-generator power plant system and radar equipment will be extended to operating position by a timing device and the vehicle will be ready for travel under control of command signals from the earth control station. This sequence of operation is illustrated in Figure 27. It should be pointed out that the vehicle shown in this illustration is an early concept and not the proposed vehicle.

B. PROPOSED VEHICLE

1. General Characteristics (Figure 28 and 29)

The proposed lunar roving vehicle consists basically of two 16 foot diameter inflatible tires connected by a dead axle. Vehicle drive motors will be located in the hub of each wheel. Vehicle track width will approximate 15 feet. The payload of the vehicle will consist of a package of electronic equipment which will be independently swung from the dead axle and the equipment will be shock mounted in two planes. The center of gravity of the package will be off-set from the axle to aid in orienting the vehicle from landing position to traveling position and to keep the package properly oriented during travel. Drive torque will be equalized through the employment of a torque reaction arm extending from the axle to the lunar surface which will trail the vehicle regardless of the direction of travel. When direction of travel is reversed the arm will rotate around the axle to its new position. A wheel with puncture proof tire will be mounted on the end of the arm in contact with the lunar surface. The vehicle will be powered by a turbo-generator operating on a simple Rankine cycle with mercury as the working fluid. Heat source for the power plant boiler will be solar energy concentrated on the boiler by a parabolic reflector. Electrical energy produced by the turbo-generator will be utilized to drive the vehicle, operate all instrumentation as required, and provide power for the instrument package cooling system. The instrument package, which is free to rotate about the vehicle axle, will be mechanically connected to the power plant structure. As it changes position on the vehicle axle under the influence of lunar gravity

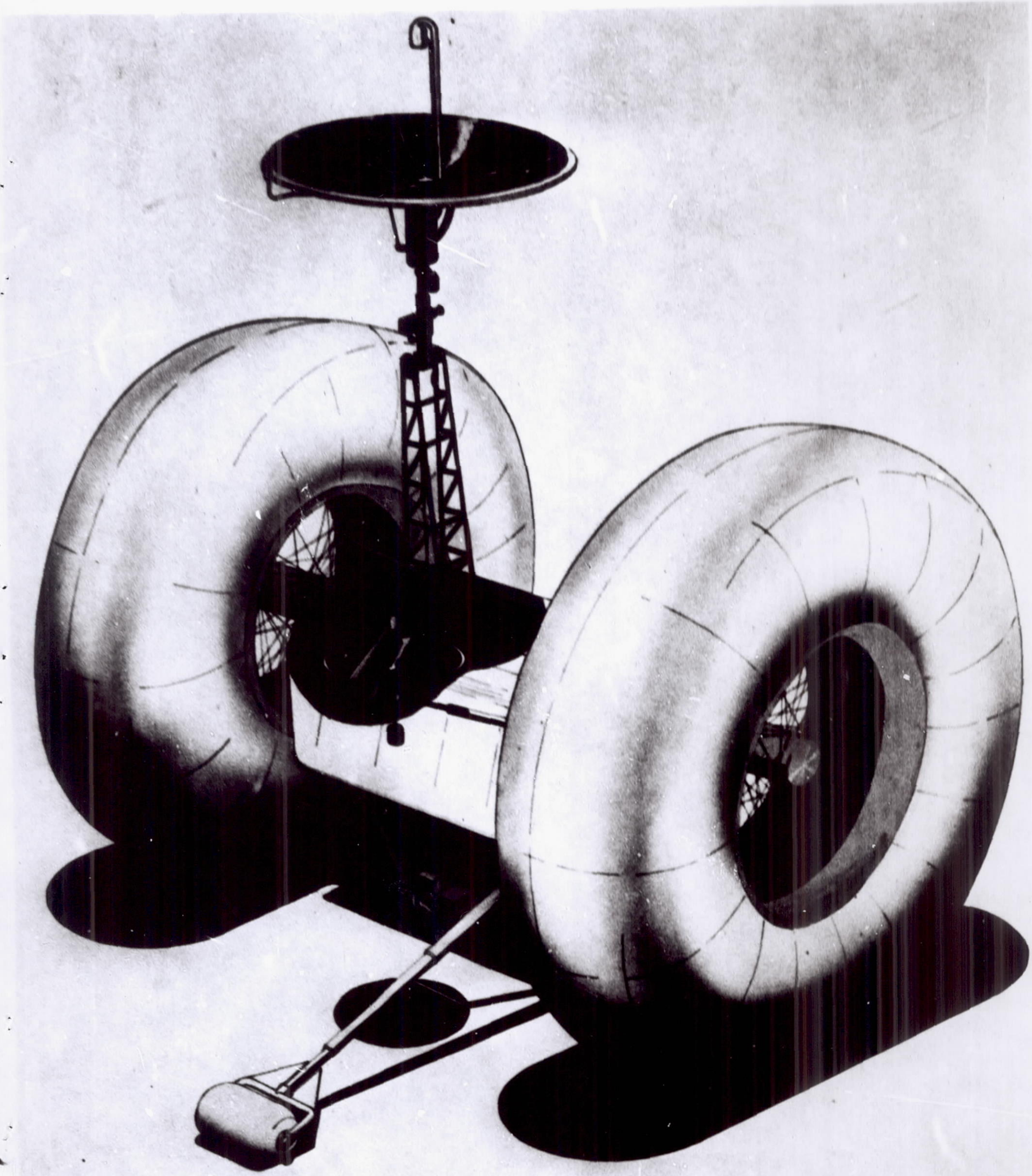
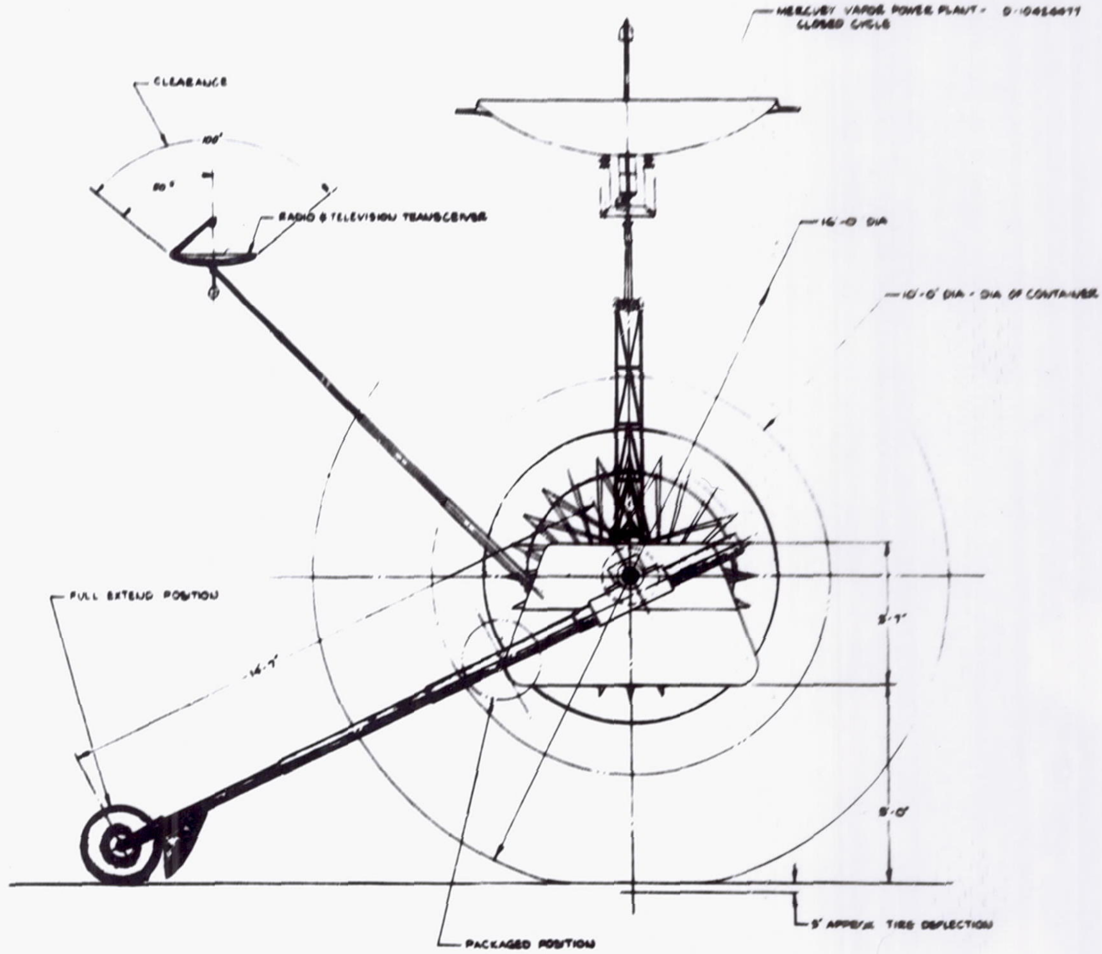
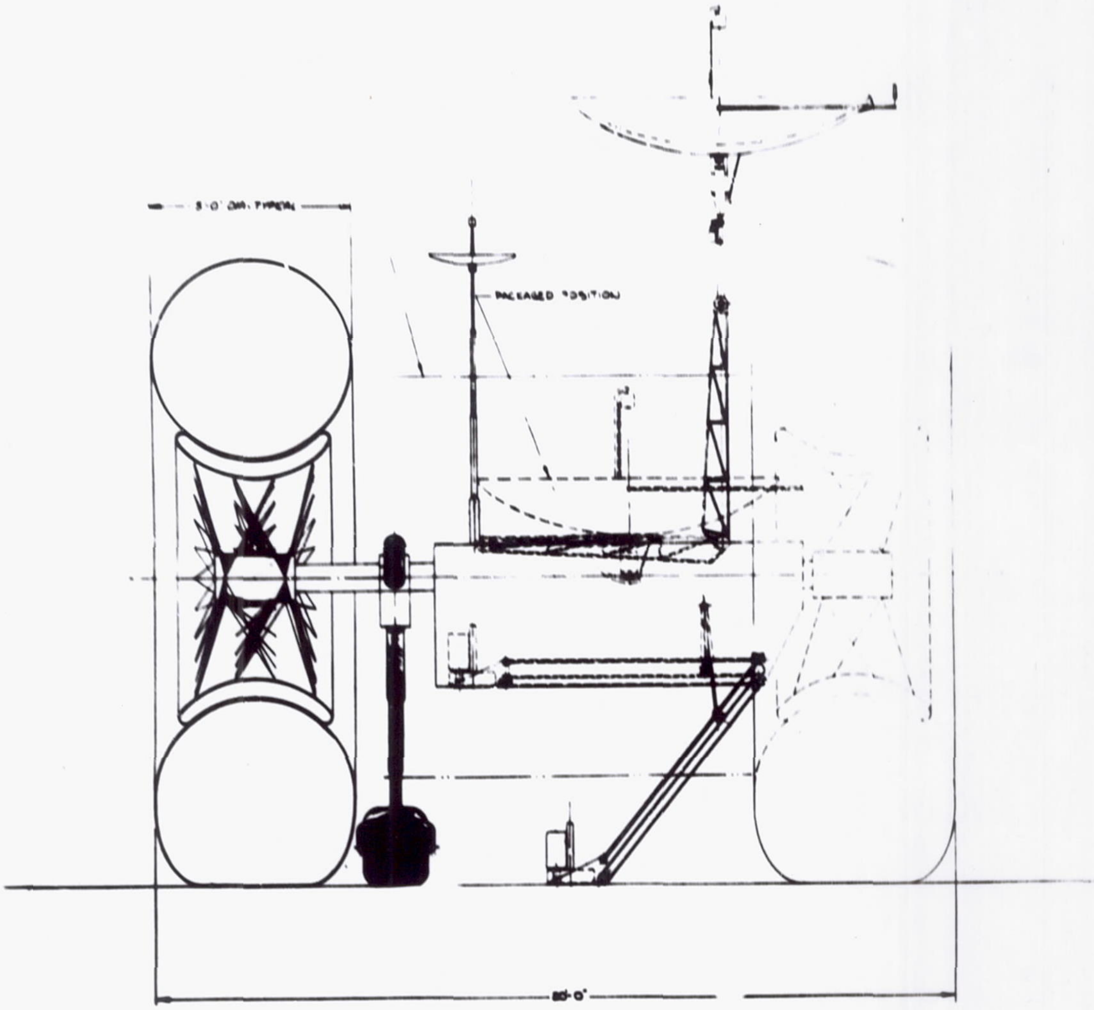


Figure 28.

LUNAR ROVING VEHICLE PROPOSAL 1



LUNAR ROVING VEHICLE (SIDE VIEW)
Figure 29A.



LUNAR ROVING VEHICLE (FRONT VIEW)
Figure 29B.

it will act as a pendulum weight to maintain the power plant in a vertical position with respect to the local surface.

The allowable weight of the payload package has now been established as 2175 pounds. This total weight will consist of the weight of the roving vehicle plus the weight of the shock absorber and final guidance and control systems which will be detached from the vehicle prior to operation. Although the total allowable weight of the roving vehicle will now be 1450 pounds, it is not considered necessary to increase the design weight to this full allowance. Allowable and design weight estimates are shown in the breakdown below. Since any increase in vehicle design weight will require more operating power and a heavier shock absorber system for landing, the vehicle design weights should be held to the minimum. All vehicle tire and power requirement calculations shown in this proposal were based on a total vehicle weight of approximately 1150 pounds (95 pounds per wheel lunar weight). The recommended design weight has now been increased to 1260 pounds as shown in the table below. The net effect of this increase would be approximately a 10% increase in vehicle power requirements or a similar decrease in vehicle design speed. Since other adjustments in design requirements will be necessary as vehicle development progresses, no change will be made in the calculations shown in this proposal.

WEIGHT BREAKDOWN

Detachable Items	Weight Allowance		Recommended Design Weight	
Separation	50		50	
Shock Absorber and Mounting	275		275	
Guidance and Control and Attitude Control	400		400	
Total Detachable Load Weight		725		725
Vehicle				
Communications System	125		125	
Power Supply (Turbine)	150		140	
Power Supply (Batteries)	125		60	
Cooling System	250		135	
Vehicle Structure, Tires Wheels, Controls, etc.	525		525	
Scientific Payload	275		275	
Total Vehicle Weight		1450		1260
Total Package Weight		2175		1985

2. Tires

a. General Considerations. The primary function of the vehicle tires will be to provide a mobility capability during the full period of operation. However, the tires will also serve a secondary function in absorbing the secondary landing shock when the vehicle topples from its vertical landing position to the horizontal traveling position. In the event a true vertical landing is not accomplished, the tires may have to absorb a part of the initial landing shock. Other major factors considered in the proposed tire design are listed below.

(1) Weight limitations (100 lbs.)

(2) Punctures and air leakage

(3) Tear, abrasion, and wear

(4) Ultra-violet effects

(5) Out-gassing effects

(6) Temperature effects

(7) Probability of micrometeorite damage

(8) Tire behavior upon encountering boulders

(9) Secondary landing effects (damping)

(10) Operating pressure and method of inflating

b. General Description. The proposed tire is basically a torus having an outer diameter of 16 feet, an inner diameter of 6 feet and a tire cross section diameter of 5 feet. The tire will be constructed of mylar polyester film or similar substance approximately 6 mils thick. An outer casing of fine mesh titanium or stainless steel wires approximately 15 mils in diameter will be embedded in this material for added strength and protection from puncture by sharp objects. Operating pressure of the tires will be approximately 1/2 psia. Prior to launch, the vehicle tires will be collapsed for storage in the missile payload section. In this collapsed condition the tire will be inflated with the amount of air necessary to expand it to full size at 1/2 psia after the payload cover is blown off prior to the lunar landing sequence. The small differential between the tire pressure and the lunar vacuum will minimize air leakage. The inner surface of the tire will be treated with a self sealing compound to prevent leakage from small punctures. Further insurance against immobilization of the

vehicle due to a flat tire can be obtained by dividing the tire into independent segments or compartments or by use of multiple tubes of small diameter covered by an outer casing. Either of these systems may be employed without exceeding the weight limitation of 100 pounds for the tires. However, other considerations such as damping of secondary shock may preclude the use of either of these schemes and final recommendations must be deferred unto further study is accomplished.

c. Damping of Secondary Landing Shock. A secondary function of the tires will be to absorb the secondary shock occurring when the vehicle topples from its landing position to its traveling position. The upper wheel of the vehicle must fall from a height of approximately 20 feet to the lunar surface. To prevent excessive bouncing, the resulting kinetic energy of this change in position must be absorbed and damped. The reaction force of the tire at any instant during impact will be the product of the area in contact multiplied by the tire pressure ($f = pa$) and the energy absorbed will be the mean of this force multiplied by the total tire deflection ($E = Fh$). The energy absorbed and stored as potential energy, less damping effects, will be converted to kinetic energy causing rebound. This process will be repeated until the rebound is less than the tire deflection. The damping effects of a simple tire would consist of tire wall flexure, heat loss due to compression of air in the tire, and compression of the lunar surface. The walls of the proposed tire are very thin and flexible, the air compression will be low and of short duration and the compression of the lunar surface may be assumed to be negligible.

Therefore, the total damping effects obtained from the simple tire will be negligible. Several methods of damping this secondary shock have been considered and are outlined below. However, no firm conclusion has been reached as to the best method to be employed.

(1) Over Inflation. Inflate the tire to an initial pressure higher than desired for operation and vent the excess air during impact. The reaction force would be less due to the reduced pressure, rebound would be less, and damping would be effected. For example, with an initial pressure of 3 psia the tire would deflect about 10 inches on impact. However, if this pressure was vented to 0.5 psia during impact the tire would deflect about 24 inches, thereby, greatly reducing the first rebound. A decrease in energy of about 1/4 on each succeeding rebound would result in approximately 12 bounces before damping would be complete.

(2) Sponge Filled Tire. Fill the tire with a very lightweight cellular material. The air in the interconnected cells of the impacted segment of the tire would be forced to the other portion during impact and would be slow in returning and damping would result. However, the volume of each tire is approximately 700 cubic feet and if a material with only 0.05 lb/cu.ft. density was used, each tire weight would be increased by 35 pounds. This additional weight cannot be tolerated with the limitation of 100 pounds for both tires.

(3) Crushable Disc. Fabricate the tire with a central internal disc of crushable material having an outer diameter equal to the outside diameter of the tire less 1 1/2 to 3 times the normal deflection of the tire as shown in Figure 30. Crushing of the material

upon impact would absorb energy and prevent rebound. Disadvantages of this system include added weight, difficulty in arranging the material within the tire to insure crushing rather than breaking under impact, and the difficulty of arranging the material within the initial diameter restriction of the missile payload.

(4) Segmented Tire. Divide the tire into a number of compartments or segments as illustrated in Figure 31. Upon impact, the pressure increase would be confined to the segment(s) in contact with the surface. With a tire pressure of 1/2 psia the total deflection upon impact would be approximately 24 inches and the tire pressure would increase about 4%. Whereas, with a six segment tire the pressure in the segment(s) impacted would increase 12 to 24% and deflection would be correspondingly less. If all segment walls were equipped with one-way vent valves operating in the same direction (clockwise or counterclockwise), this excess pressure would be vented to adjoining sections in turn and the gradual equalization of pressure in all sections would achieve a damping effect. This scheme would nullify the advantage of a compartmented tire in preventing completely flat tires because all sections would gradually vent to lunar vacuum through the punctured tire.

d. Tire Deflection Calculations

(1) Level Terrain (Figure 32). When a round flexible tire under low pressure is rolling on a level surface it will deflect and the area in contact with the surface will become elliptical in shape. This area will be equal to the total load on the tire divided by the tire pressure ($A = \frac{F}{P}$). For purposes of tire calculations, the traveling weight of the roving vehicle is assumed to be approximately 1150

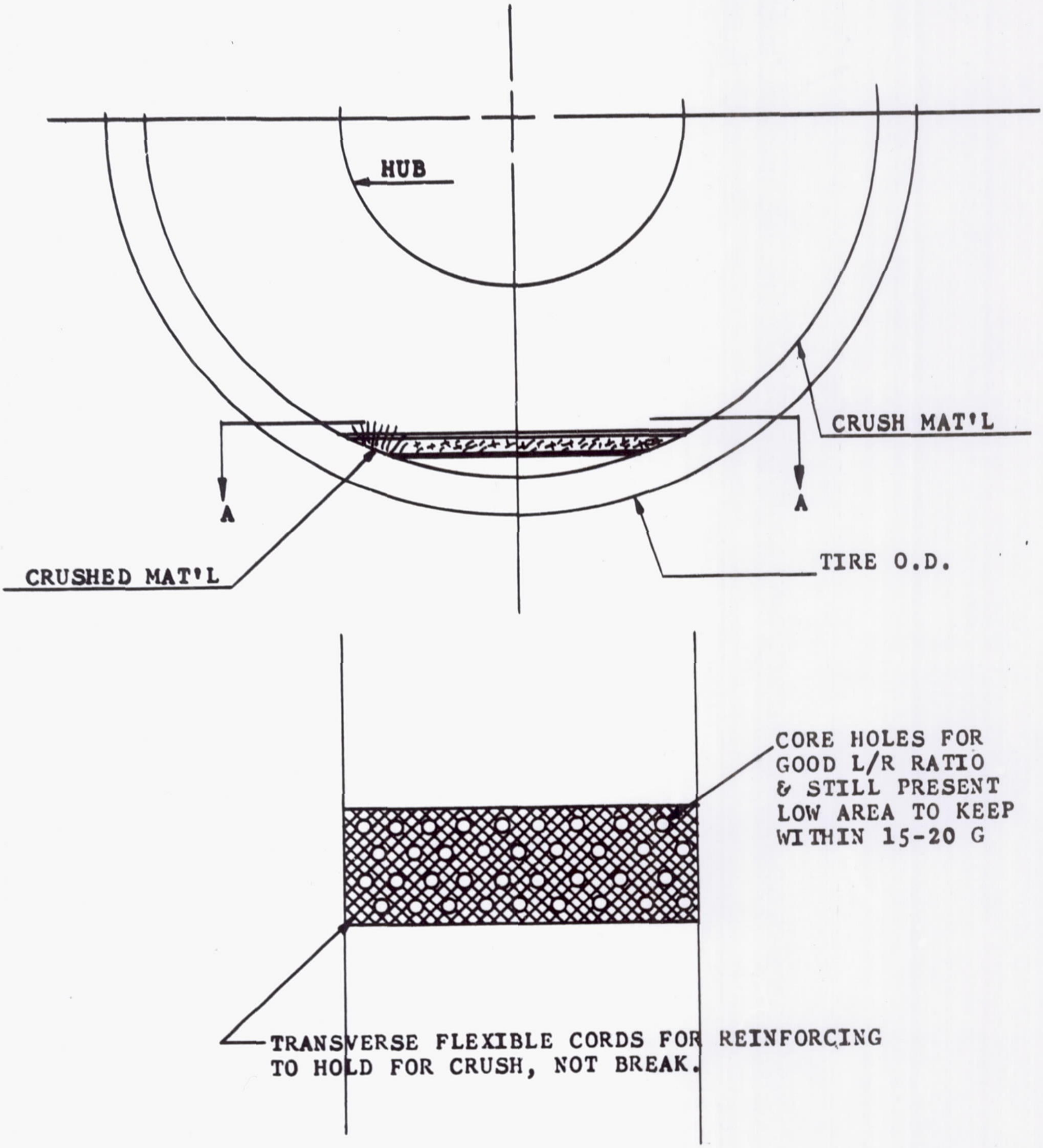


FIG. 30. CENTRAL DISC, ENERGY ABSORBER

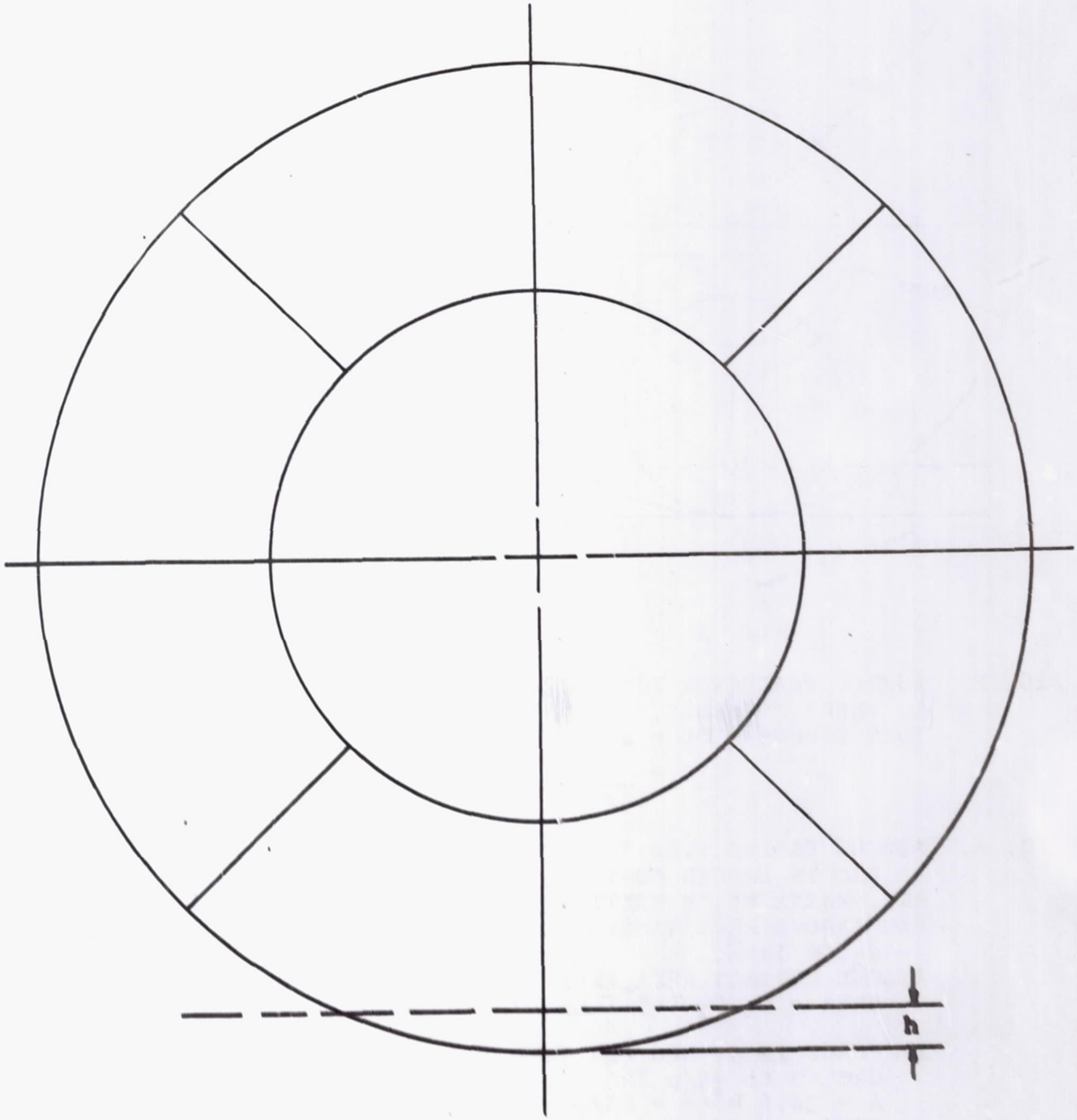


FIG.31. COMPARTMENTED TIRE

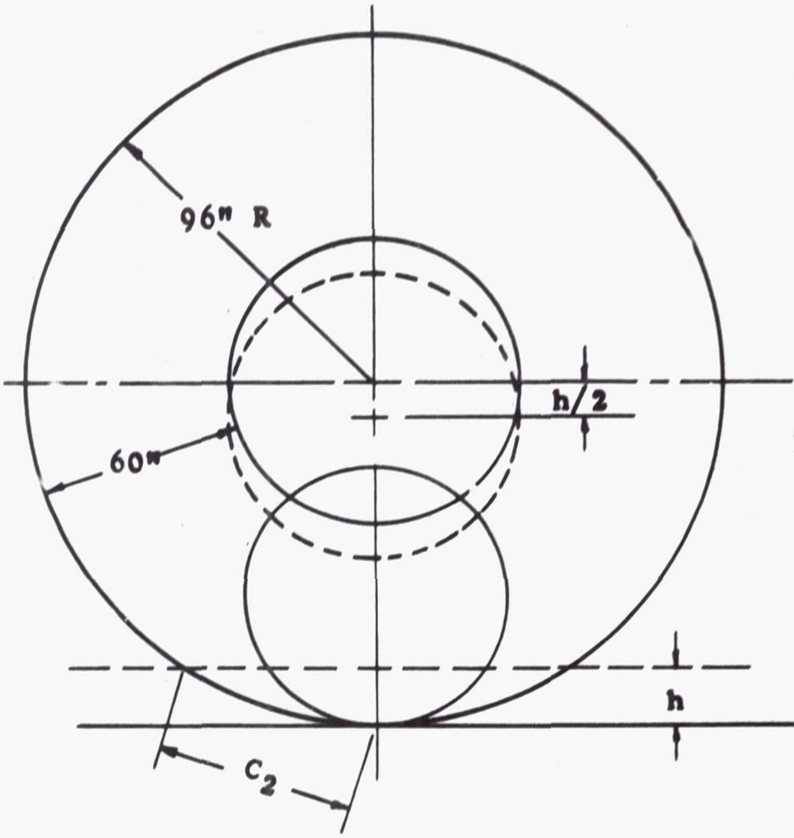


FIG. A

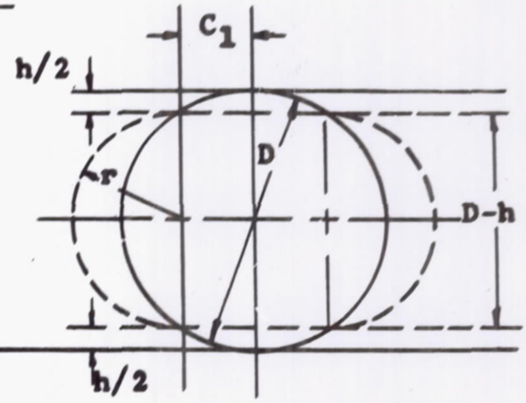


FIG. B

FIG. B. ASSUME PERIMETER INELASTIC AND OF CIRCULAR CROSS SECTION WHERE POSSIBLE.

THEN CIRCUM $= \pi D = 4C_1 + \pi(D-h)$

$$C_1 = \frac{\pi}{4}h$$

FIG. A. ASSUME CASING TIED TO RIM AND KEEPS CIRCULAR SHAPE WITH ARC IN GROUND CONTACT.

REF. MARKS HB, 5 EDITION, SEGMENTS OF CIRCLES, PP 34-35.

FROM ABOVE REF. TABULATED VALUES WERE EMPLOYED TO DERIVE $C_2 \cong 14\sqrt{h}$.

ASSUME CONTACT AREA ELLIPTICAL WITH C_1 AND C_2 AXIS.

AREA $= C_1 C_2 \pi = \frac{\pi}{4}h \cdot 14\sqrt{h} \cdot \pi = 34.6h^{3/2}$

IF FORCE IS 95 LBS PER TIRE THE AREA PRESENTED TO THE GROUND IS 95/p IN^2

$A = 34.6\ h^{3/2} = 95/p$

THEN $p = 95/34.6\ h^{-3/2}$

OR $h = \left(\frac{2.7}{p}\right)^{2/3}$

FIG. 32. TIRE DEFLECTION, NORMAL TERRAIN

pounds (1750 pounds less 350 pounds guidance and control and 250 pounds deceleration equipment) which will give an individual tire load of approximately 95 pounds (lunar). Derivation of the following formulas for estimation of the amount of deflection (h) is shown in Figure 32.

$$A = \frac{F}{P} = 34.6\ h^{3/2}$$

$$h = \left(\frac{2.7}{P}\right)^{2/3}$$

Where:

A = area in contact with ground (sq. in.)

h = tire deflection (in.)

P = tire pressure (psia)

F = total load on tire (force)

(2) Crossing Boulders (Figure 33). The work required to roll the tire over a boulder is a function of the boulder size and the tire pressure. Derivation of formulas to be used in estimating work required was based on the following assumptions as illustrated in Figure 33.

(a) Spherical boulders

(b) Non-elastic but fully flexible tire casing

(c) With tire to boulder contact, that portion of the tire in contact would assume a circular cross section with radius r.

(d) When boulders are large, the contact area supporting the tire will be elliptical and pressure will remain almost constant.

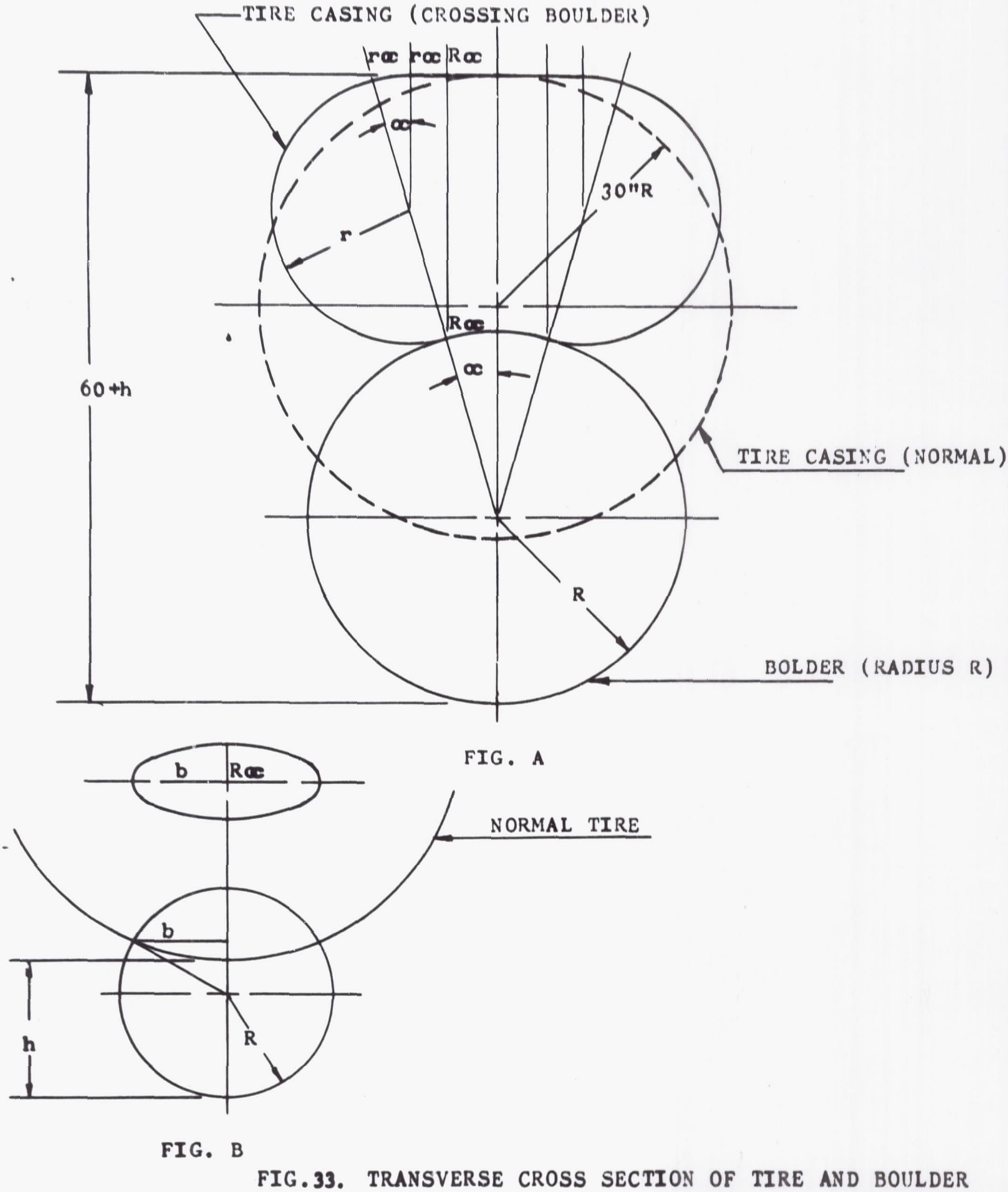


FIG. 33. TRANSVERSE CROSS SECTION OF TIRE AND BOULDER

With these assumed conditions the following formulas were derived for determining resistance to travel over large boulders:

$$\alpha = \frac{\pi}{2} \quad \frac{(30 - r)}{(R + r)}$$

$$h = r + R\ (R + r)\ \cos \alpha\ - 60$$

$$b = (2\ Rh - h^2)\ ^{1/2}$$

$$A = \pi\, b\, R\, \alpha$$

$$p = \frac{95}{A}$$

Where:

R = radius of boulder (in)

r = radius of free portion of tire (in)

α = angle from center of boulder subtended by transverse contact of tire with boulder (radians)

h = rise of axle caused by travel over boulder (in)

A = contact area of tire with boulder (sq.in.)

p = tire pressure (psia)

b = semi-chord of boulder in longitudinal direction of intersection of tire perimeter (in)

These formulas were used for computing the data for the tire pressure - deflection curves shown in Figure 34. This graph indicates the amount of rise of the axle (h) and cousequently the work performed, (w = Fh) when traveling over large boulders of varying diameter with different tire pressures. It should be emphasized that the work required will be a function of the bounder size and tire pressure and that lower tire pressures will greatly reduce the work required to cross any given boulder size. The total work required will be performed over a period from initial contact until the vehicle axle is directly

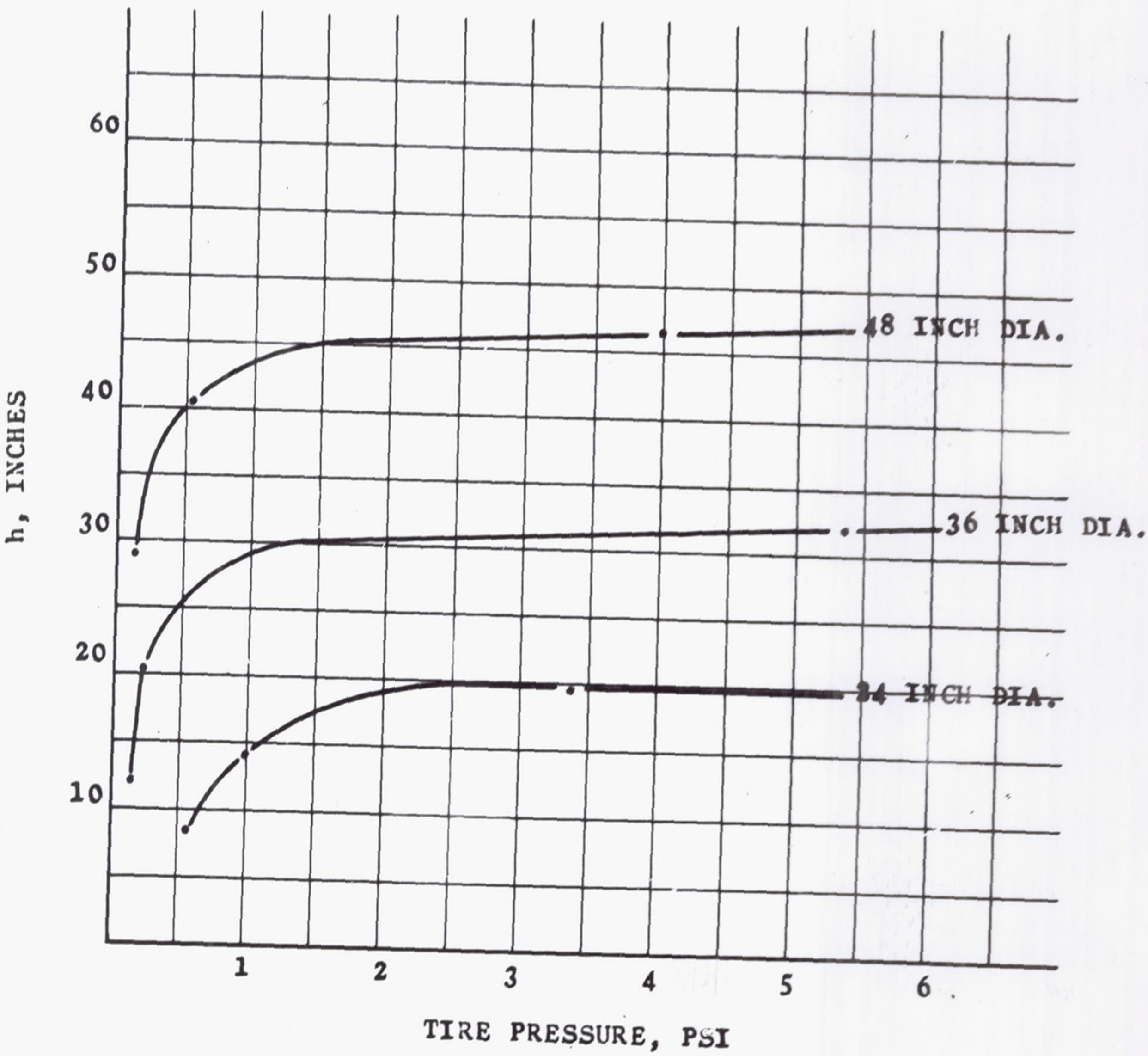


FIG. 34. VERTICAL AXEL DISPLACEMENT VS TIRE PRESSURE FOR VARIOUS BOULDER DIAMETERS

over the center of the boulder. Boulders are considered large in this discussion when their cross-section is large enough to entirely support the tire. Smaller boulders will be largely enclosed by the tire and will require some additional ground area for tire support. The work required to travel over these types is small as the actual lift of the axle is a small fraction of the boulder diameter.

(3) Crossing Crevasses. In roving over the lunar surface, the vehicle will at times be required to negotiate crevasses. The work required will depend on the width of the crevasse and tire pressure.

With the vehicle axle over the center of the crevasse, tire depression at the edge of the crevasse will be:

$$h = \frac{2w}{p^2}$$

Where:

h = tire depression

w = width of crevasse

p = tire pressure

When the axle is directly over the edge of the crevasse,

$$h = \frac{(5.5)}{p}^{3/2}$$

and reaches the normal

$$h = \frac{(2.7)}{p}^{3/2}$$

after it travels a distance equal to the major axis of the contact ellipse ($14 \sqrt{h}$).

3. Drive Power Requirements

Determination of the maximum power requirements for mobilization of the roving vehicle was based on the following assumptions:

a. Vehicle speed of 3 mph

b. Zero tire deflection, hard tire

c. Friction loss due to tire-to-surface contact is zero. (This includes tire material hysteresis loss and surface deformation).

d. Normal travel will be over flat surface with maximum slope of 15°. Negotiation of 3-4 foot diameter boulders will be required. A boulder of this nominal size could be encountered on a 15° slope.

Even though it has been shown that less power is required to cross a boulder with a soft (low pressure) tire than with a hard tire, a hard tire analysis is used here to establish a maximum power requirement. Computation of exact power requirements for soft tires requires the use of specific data for each situation. Since specific data required is not known in all cases, the assumption of a hard tire was made for calculation purposes. This will establish a maximum power requirement for the most adverse circumstances and will provide a safety factor for normal operations. Any boulder or crevasse can be converted into an equivalent slope for a hard tire. In the case of boulders, it was assumed that they would be approximately spherical and on top of the surface. The maximum instantaneous torque required will occur at the first point of contact and will then decrease to zero when the vehicle axle is directly over the top of the boulder. With a soft tire, the increase in torque will be gradual due to tire deflection and the maximum instantaneous torque will decrease due to a

decrease in equivalent slope. These two conditions are illustrated in Figure 35. The table shown with Figure 36 was compiled to show the drive torque requirements for the design slope of 15° and the equivalent slopes for boulders and ditches of varying diameters and widths respectively. Also shown in this table is the required torque arm reaction for the various conditions. The graph of Figure 37 shows the relationship between slope and torque requirements per wheel. The graph of Figure 38 indicates the speed versus torque requirements for a family of constant power curves. These graphs on vehicle mobility characteristics were based on an assumed vehicle load of 93 pounds (lunar) per wheel as stated for tire calculations. Total continuous power requirements for movement at 3 miles per hour on a 15 degree slope was determined to be 0.38 horsepower (0.283 KW) as shown below.

Torque per wheel on 15° slope	=	192 ft. lbs (From Fig. 37)
		2
Total torque	=	384 ft. lbs.

From Figure 38, at a torque of 384 ft. lbs. and a speed of 3 miles per hour we find that 0.38 hp is required. In the event the vehicle encounters a boulder or ditch of the specified sizes, the vehicle speed will be reduced accordingly, and since power delivered to the wheel will remain constant, the torque will increase to negotiate the obstacle. For example, if a boulder 3 feet in diameter is encountered by one wheel on a 15° slope the total torque required will be 934 ft. lbs. (550 + 384) and the vehicle speed will be reduced to 1.1 mph (From Fig. 38).

4. Drive and Control System

It is proposed that the vehicle be driven by two separate electrical motors in order to accomplish the simple control system desired. One motor, along with a planetary reduction gear will be housed inside the axle at each wheel hub. The motors will be driven by AC current from the turbo-generator of the main power plant. Vehicle control will be achieved by using servo units attached to each motor. When both servos are turning at the same speed their electrical output will be the same and will cancel out. When their speeds are different, their different outputs, when added, will have a net output which will operate a control mechanism to bring the two wheels to the same speed. External signals can be fed to the control mechanism to maneuver the vehicle. With a vehicle speed of only 3 mph, control of the vehicle can be simple yet satisfactory. Time delay affects of radio and TV signal travel from the moon to earth and back (approximately 2.6 seconds) tend to make precision control of the vehicle difficult. The value of a complex system, such as speed control and variable turning rate controls, is doubtful largely because of this time delay effect. A simple "on", "off", "reverse" control for each wheel can be used with no provision for speed control. Maneuver of the vehicle can be accomplished by stopping one wheel while the other continues to turn or by stopping both wheels and reversing one. The angular change of vehicle direction will be directly proportional to how long one wheel is stopped. Other important advantages of simple controls are decreased weight and increased reliability.

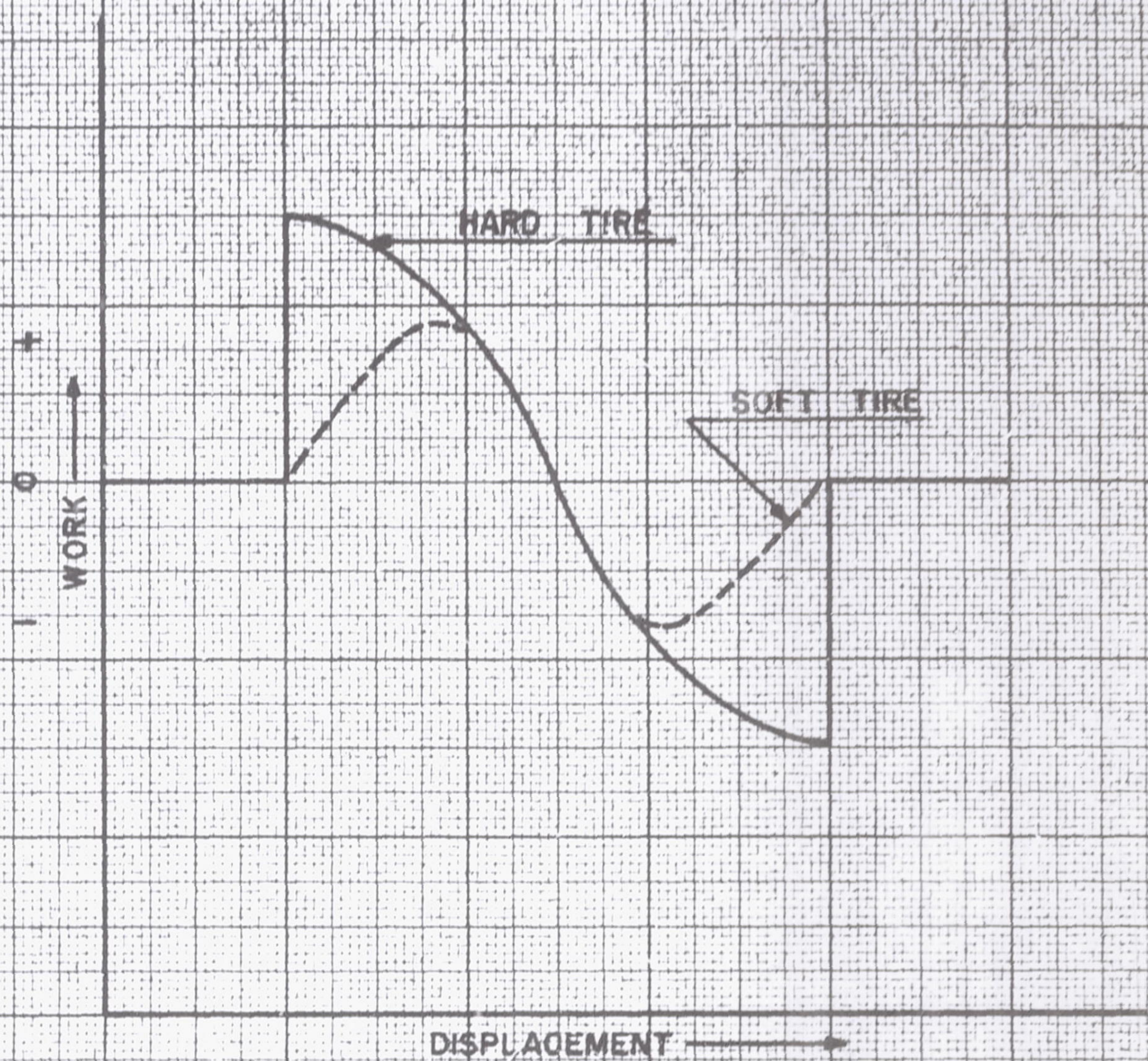
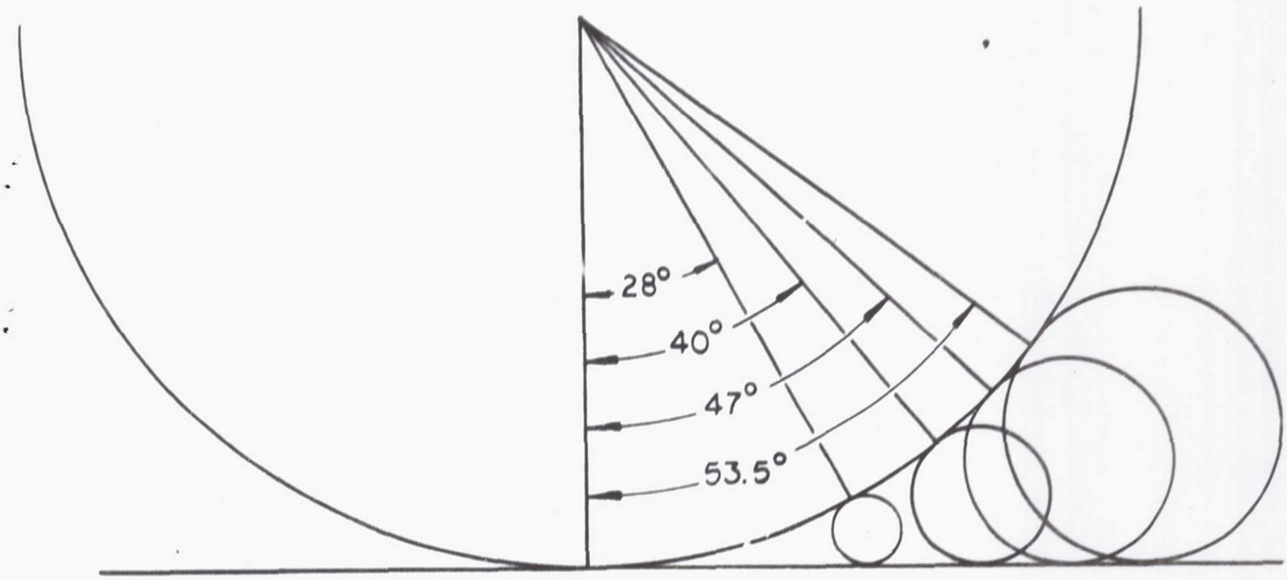


FIG. 35. WORK, HARD VS. SOFT TIRE



BOULDER DIA. FT.	DITCH WIDTH	EQUIV. SLOPE	TORQUE FT. #	TORQUE ARM REACTION LBS.
		15° *	192	12.8
1.0	7.50	28°	350	23.3
2.0	10.00	40°	480	32.0
3.0	11.75	47°	550	36.7
4.0	13.00**	53.5°	600	40.0

* DESIGN POINT

** DUE TO TIRE DEFLECTION A DITCH OF THIS WIDTH PROBABLY COULD NOT BE NEGOTIATED

FIG 36. BOULDER SIZE - EQUIVALENT SLOPE VS TORQUE REQD

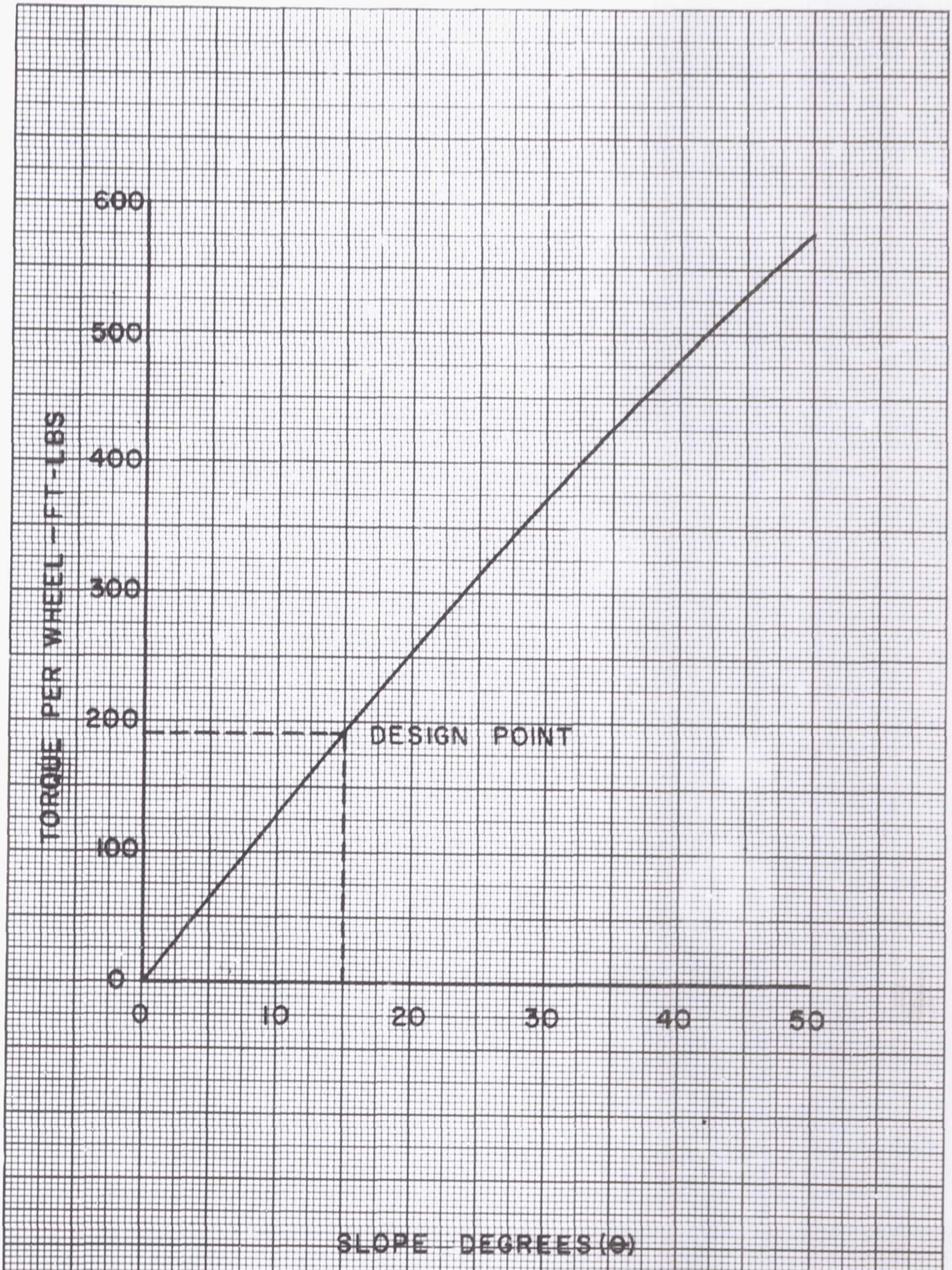


FIG. 37. SLOPE VS. TORQUE PER WHEEL

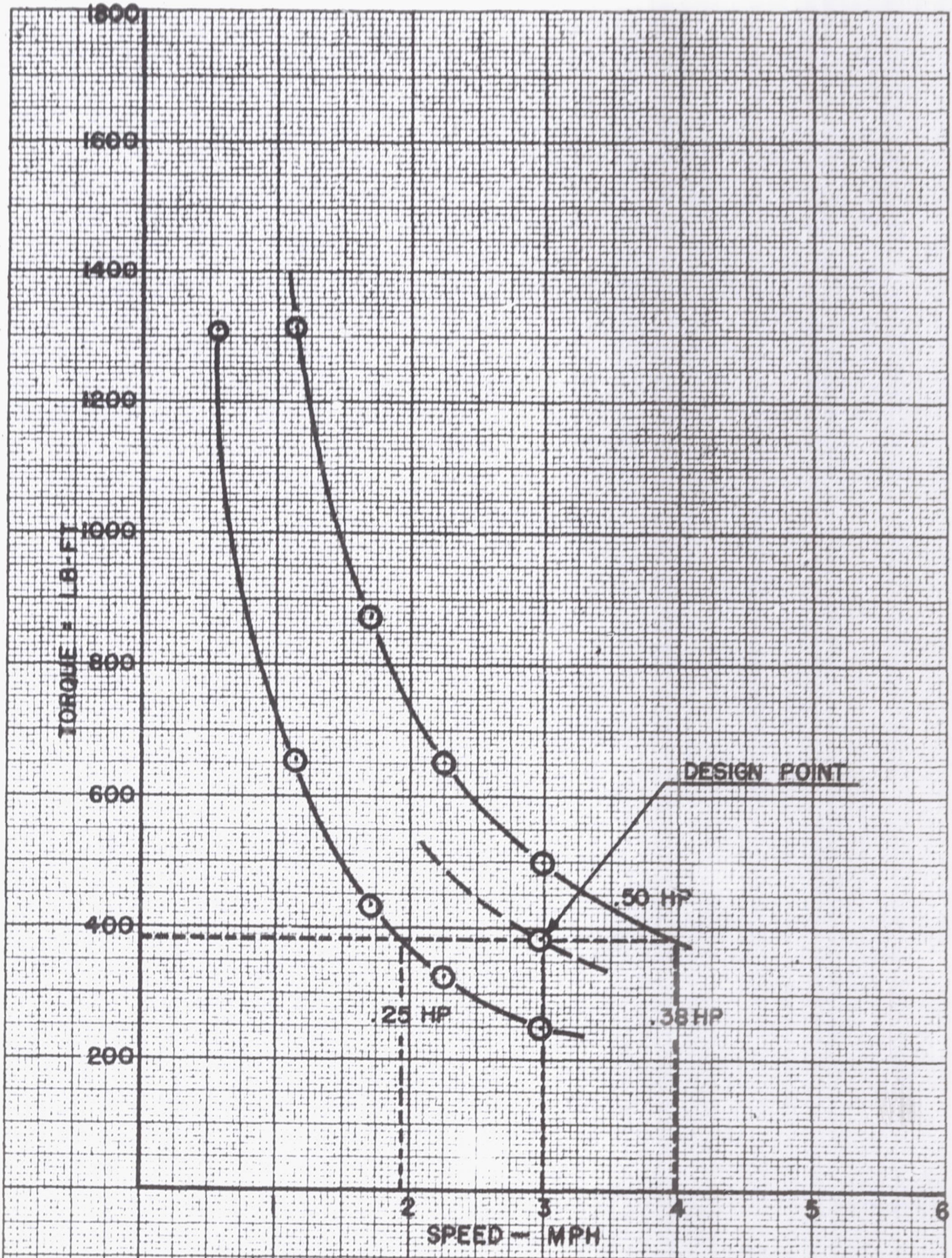


FIG.38. VEHICLE TORQUE VS. SPEED FOR CONSTANT POWER CURVES

C. ALTERNATE VEHICLE

The vehicle shown in Figures 39 and 40 was envisioned during the early stages of the design studies but was later replaced by the proposed vehicle discussed in paragraph B. above. This vehicle resembles the proposed vehicle in general configuration but has two torque arms instead of one. One arm would be employed in forward drive and the other in reverse. In this version, a TV scanning camera and core drill would be mounted on each torque arm instead of in the instrument compartment as planned for the proposed vehicle. Torque reaction and instrument operation would change from one arm to the other as the vehicle changed direction. This vehicle was to be powered by an open-cycle turbine driven electric generator system using a storable propellant such as hydrazine. The instrumentation was to be powered by a solar cell system. This concept of the lunar roving vehicle was discarded in favor of the proposed vehicle after a more complete study was made.

D. INSTRUMENT COMPARTMENT COOLING SYSTEM

1. General

The lunar vehicle payload is considered as the instrumentation package which it carries to perform the desired investigation of the lunar environment. The proposed apparatus assembly includes:

a. Television System

b. Gravimeter

c. Seismic Apparatus

d. Thermal Measrrement Devices

e. Sample Collecting Equipment

f. X-Ray Fluorescence and Diffraction Equipment

g. Gamma Ray Scientillation Spectrometer

This apparatus will be housed in an instrument package swung beneath the vehicle axle in the traveling position. All instruments requiring cooling to maintain suitable operating temperatures will be housed in an insulated and refrigerated compartment within the package. Some of the apparatus, such as the sample collector which does not require a controlled operating temperature, will be housed in an uncooled compartment of the package during travel and extended on an articulated arm for operation. Required size of the package cannot be determined until details of instrumentation and cooling apparatus have been established and final arrangement within the package is decided. Peak load power requirements for instrument operation must also be determined after an operating sequence is established.

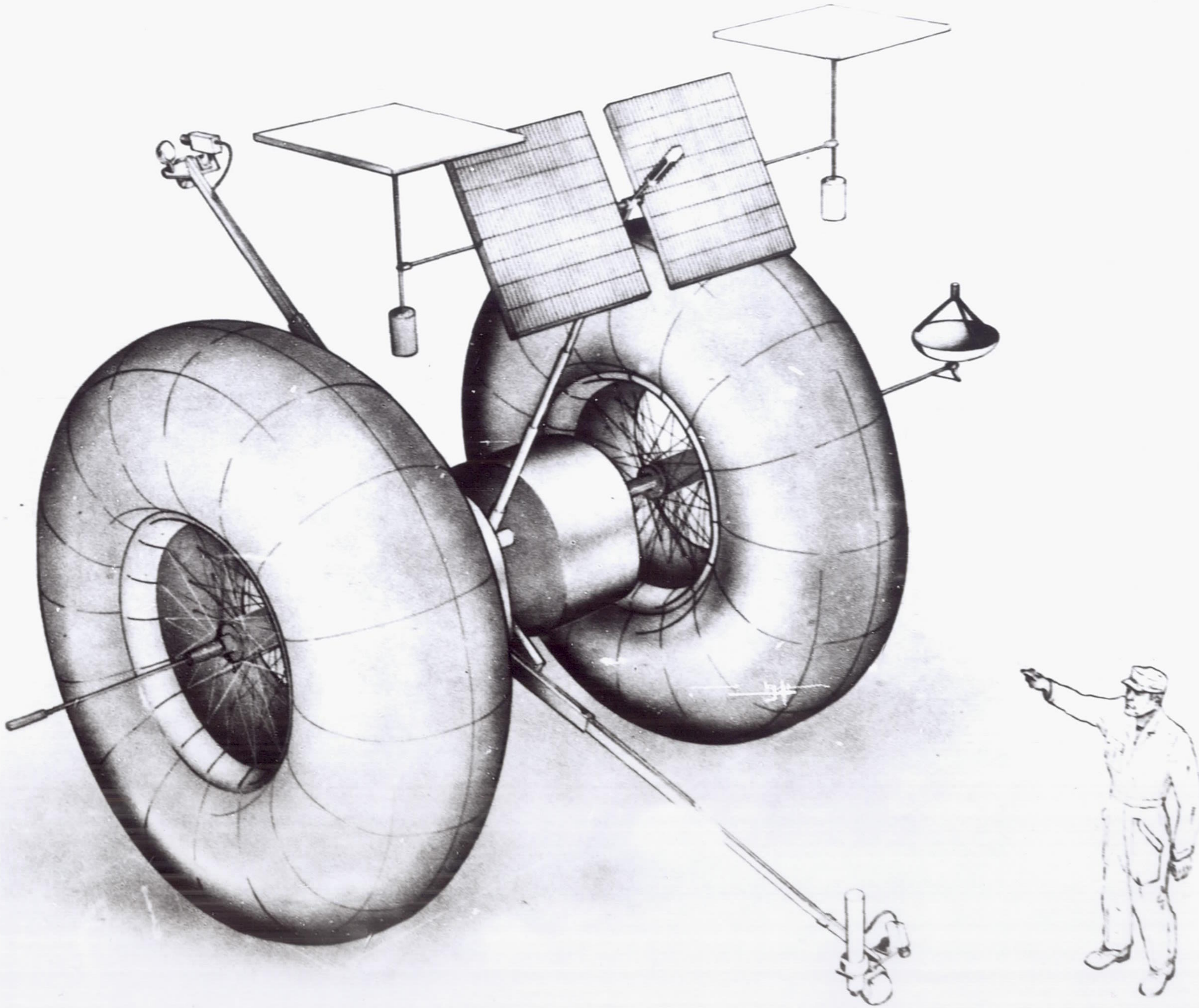


Figure 39. **LUNAR ROVING VEHICLE** (ALTERNATE)

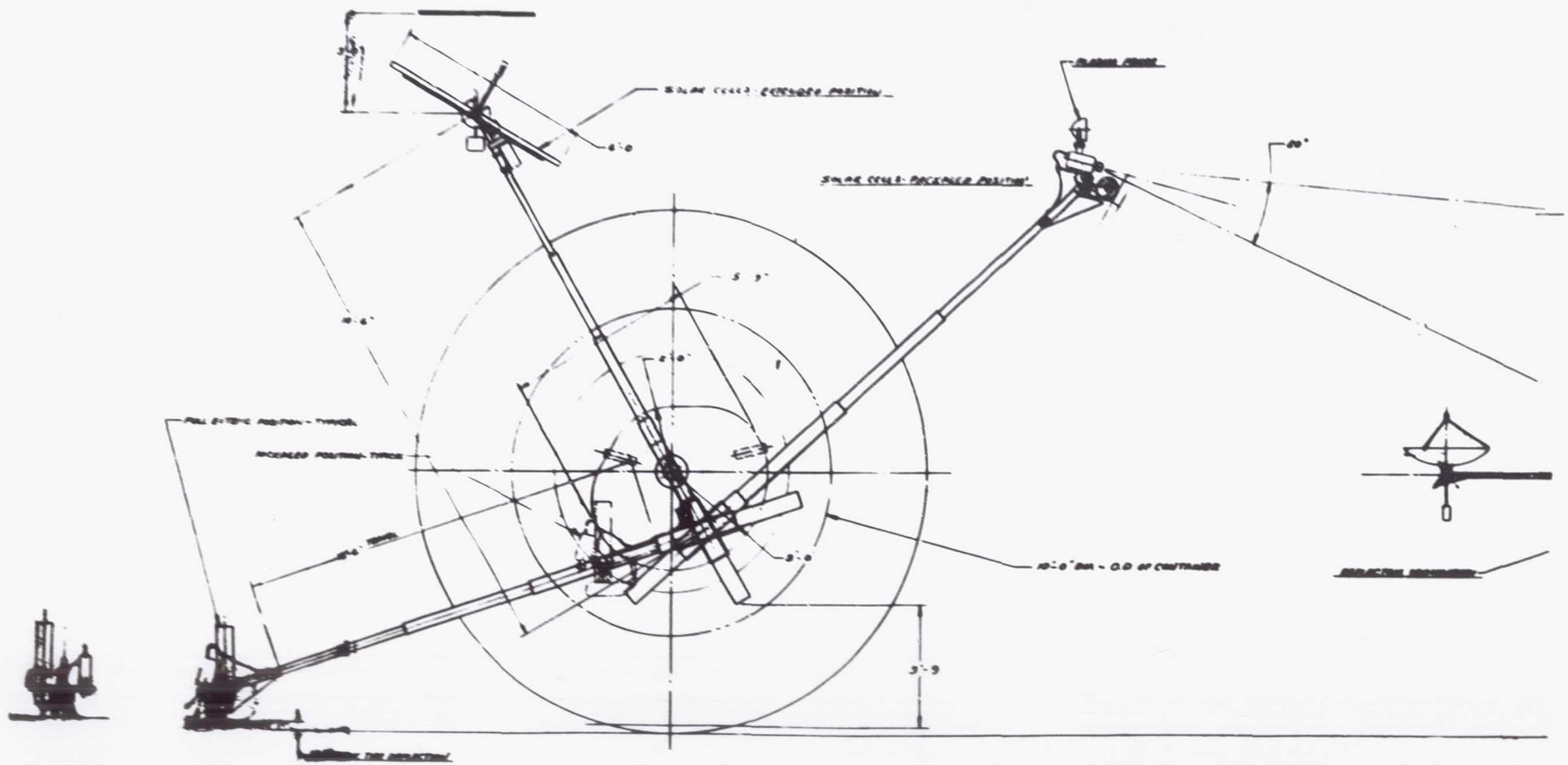


Figure 40A. LUNAR ROVING VEHICLE - ALTERNATE (SIDE VIEW)

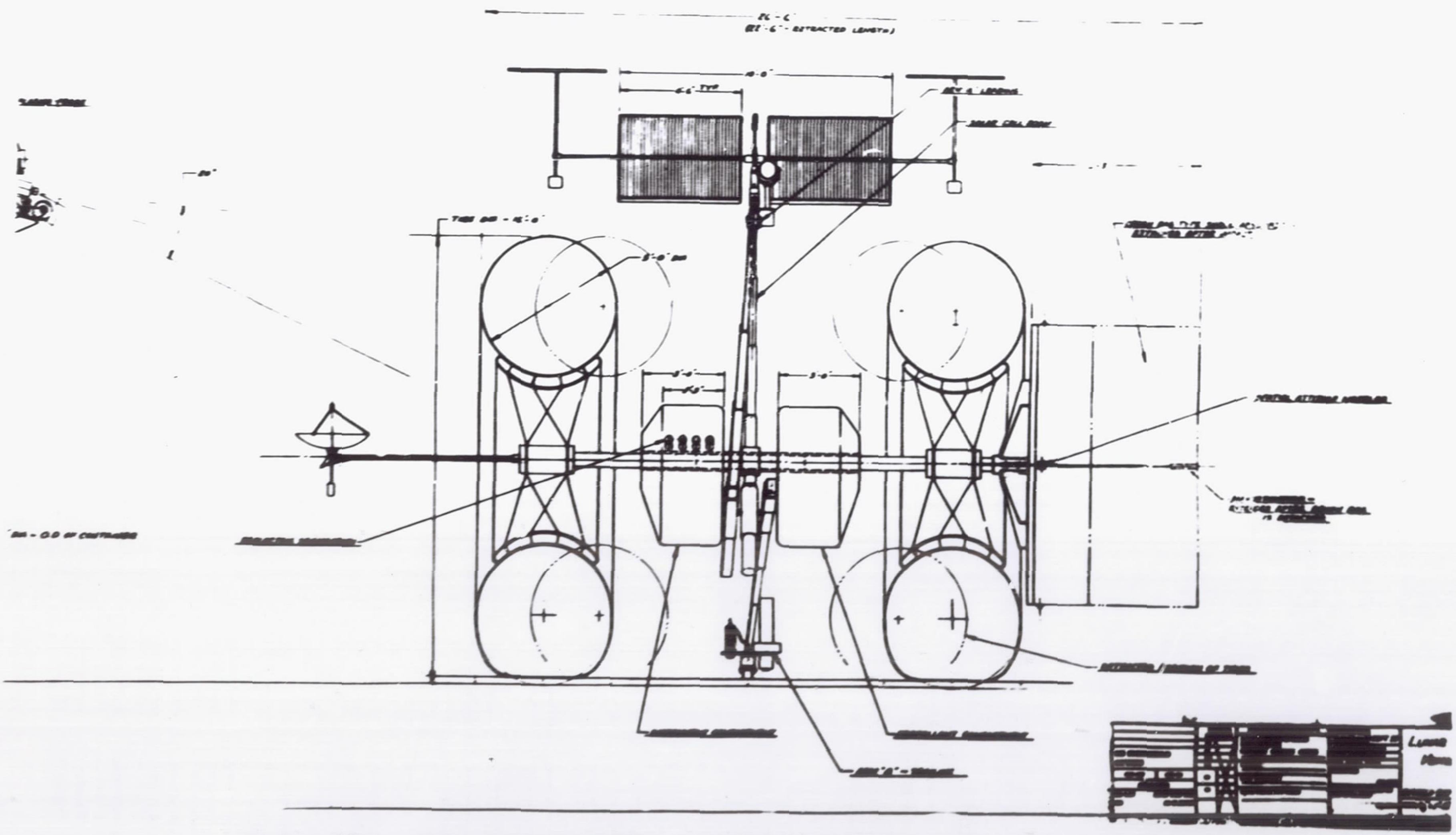


Figure 40B. LUNAR ROVING VEHICLE - ALTERNATE (FRONT VIEW)

2. <u>Characteristics</u> (Figure 41)

Instrument compartment cooling will be accomplished by a refrigeration unit operating on a reversed Brayton Cycle. The compressor will be driven by an electric motor which will be operated by power from the main power plant turbo-generator. After compression, the working medium will be routed to a radiator where heat will be dissipated at constant pressure. The design operating temperature of the radiator will be above the lunar surface temperature. Consequently, the radiator can emit heat on all surfaces and will not require constant orientation as the vehicle changes position. The working medium will be routed from the radiator to an expansion turbine where a considerable additional temperature drop will occur. The medium, now at its low temperature, will be injected directly into the instrument compartment for cooling. The work gained in the expansion turbine will be utilized to aid in driving the compressor which will reduce the net external power requirement to operate the system.

Using Hydrogen as a working medium, this system will provide the instrument compartment with a gaseous atmosphere at a pressure of 5 psia. The gas will enter the compartment at 38°F and leave at 140°F. Instruments can be placed in the compartment so that those requiring the lowest operating temperature will be near the inlet side of the compartment and those with higher tolerable operating temperature will be near the warmer outlet side. A system schematic and temperature entropy chart based on Hydrogen is shown in Figure 41.

With the cooling load of 6.137 BTU per minute the power requirement for the system will be 0.469 hp (0.35 KW). Preliminary

calculations show that other working mediums such as air or nitrogen will result in approximately the same power requirements. Final choice of a working medium will be governed by more detailed study of such characteristics as leakage susceptibility, chemical reactions within the instrument compartment and density as related to fluid friction losses in lines and rotating components.

3. Summary of Calculations (See Appendix C)

Since all factors affecting the cooling system are not known at this time it was necessary to make certain assumptions. The load and power requirements stated above were determined on the basis of the following assumptions which are believed to be reasonable enough to give indicative results.

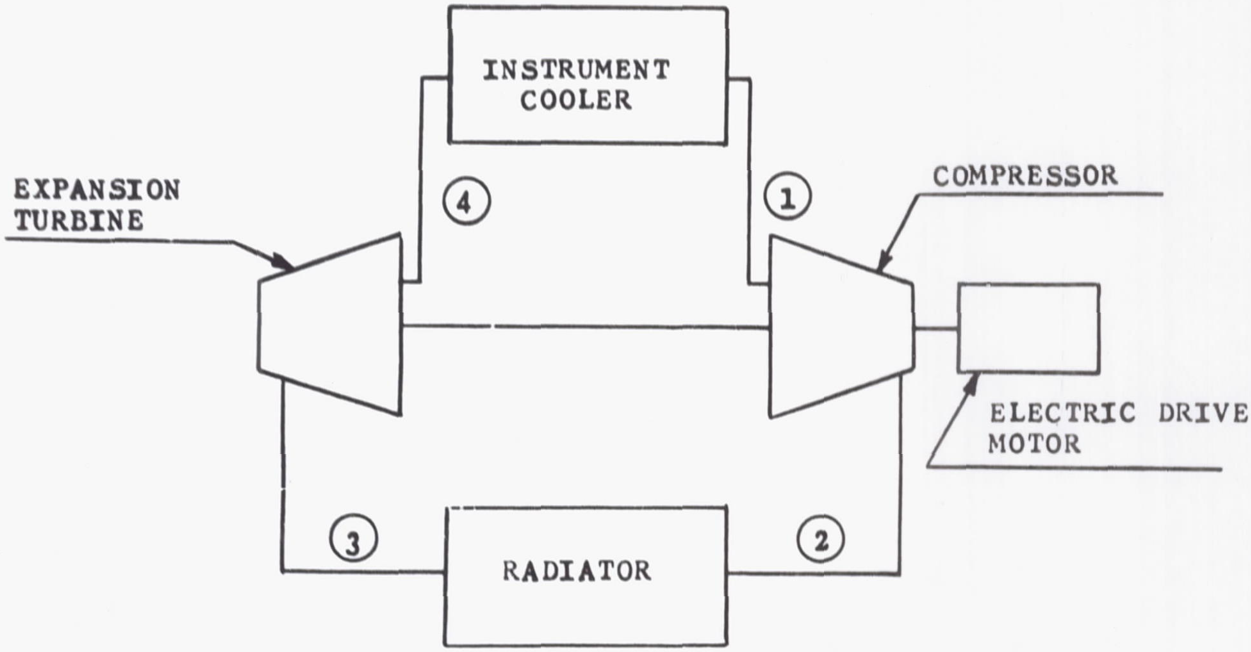
a. Continuous internal heat load from instrument operation 100 watts (5.688 BTU/min).

b. Instrument compartment size 3' x 2' x 4'; 52 sq. ft. external surface.

c. Linde S-14 type insulation to be used.

d. Weight of compartment 300 lbs.

The compartment may be supported within the payload package by structural members or by the insulating material. With either method of support, the primary considerations will be strength and rate of heat transfer into the compartment through the supporting members. To illustrate the wide difference in heat transfer, when using metallic supports and when using the insulation as supports, calculations were made on a support system using piano wire suspension and Linde S-14



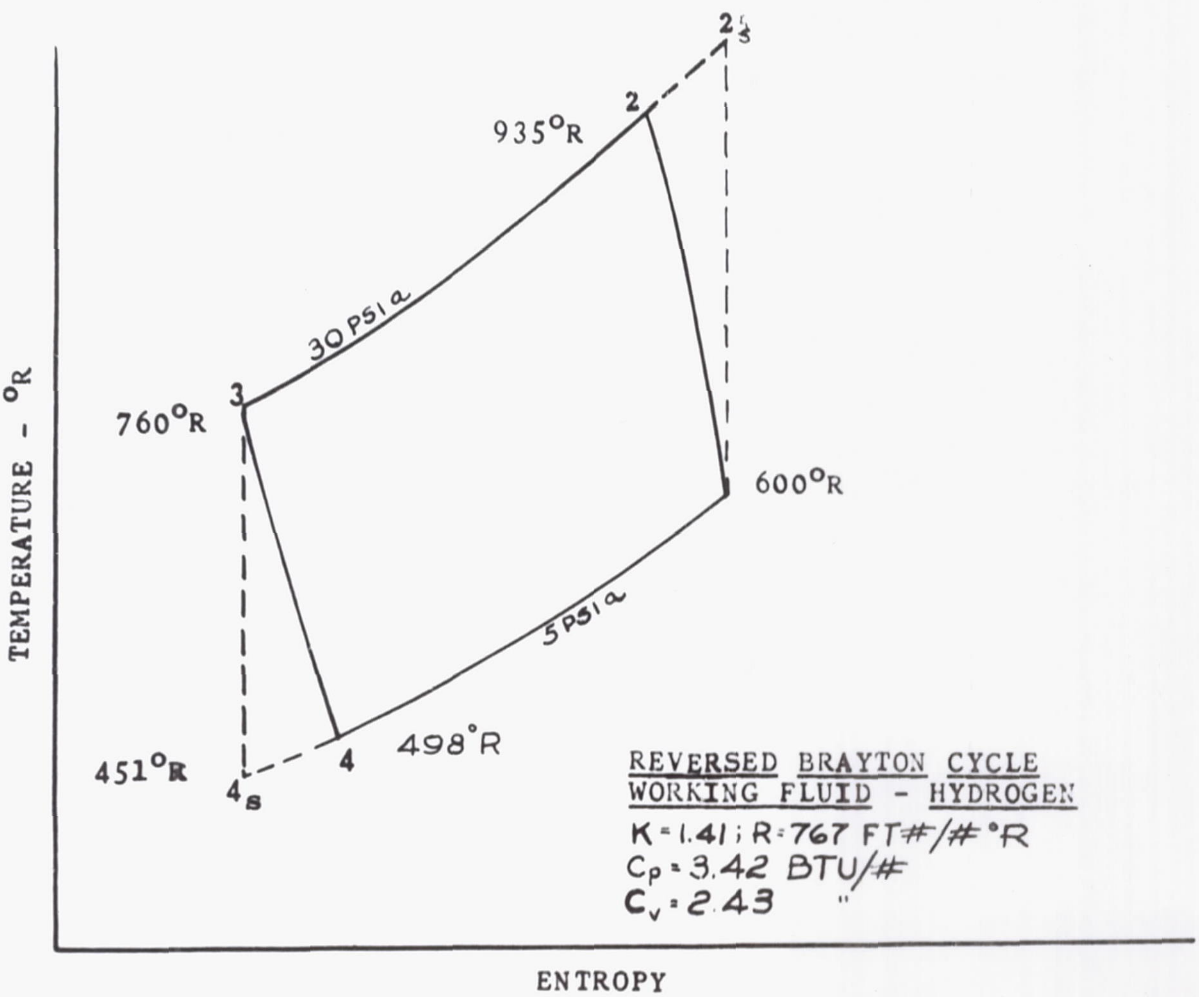


FIG. 41. INSTRUMENT COMPARTMENT, COOLING SYSTEM

insulation. Results showed that heat transfer using piano wire would be 0.571 BTU/min and using the Linde insulation it would be 0.174 BTU/min. Therefore, it is proposed to use the Linde S-14 insulation as supporting members since it has adequate strength for this application.

Two other sources of heat transfer into the compartment must be considered. These are the ducts used to circulate the coolant medium to and from the compartment and the electrical lines entering the compartment. Since the coolant transfer lines will also be insulated they will not greatly influence the heat transfer into the compartment and may be disregarded. However, calculations show that electrical lines entering the compartment will result in a heat transfer of 0.275 BTU/min.

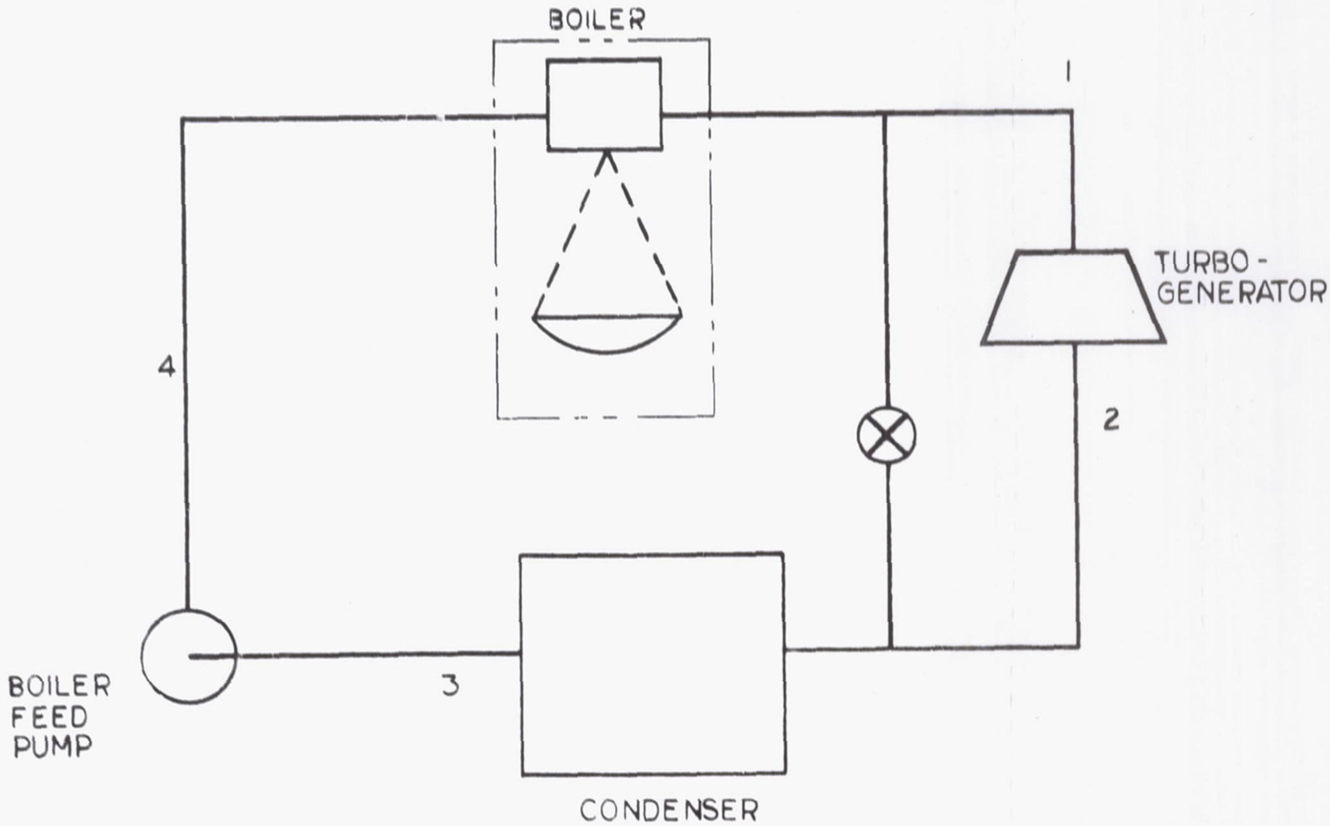
Therefore, the total heat load on the cooling system will be:

Internal load (100 Watts)	5.688 BTU/min
Load through insulation	0.174
Load through electrical wires	0.275
Total Load	6.137 BTU/min.

E. POWER PLANTS

1. Proposed Power Plant

a. Characteristics (Figures 42, 43 and 44). The proposed power plant for the lunar roving vehicle is based on a closed cycle turbo-generator utilizing a simple Rankin Cycle with mercury vapor as the working fluid. This type power plant may be utilized for either of the two roving vehicles considered in this report. The operating cycle is shown schematically in Figure 42 and a typical temperature-entropy (T-S) diagram is shown in Figure 43. Solar energy is utilized to fire the boiler which delivers saturated mercury vapor to the turbine at the required temperature and pressure. The turbine exhaust is then condensed and returned to the boiler by the boiler feed pump to complete the cycle. The total weight of this power plant, including the working fluid, will be 100-140 pounds. The selection of a boiler operating temperature and pressure is influenced by cycle efficiency, heat loss, engine and boiler metallurgical limits, wall thickness considerations as related to heat transfer and weight, and quality of the mercury vapor as it passes through the engine. As the boiler pressure is increased with saturated vapor conditions, the heat transfer losses by radiation increase and vapor quality leaving the engine decreases as shown in Figure 44. The weight of engine housing, boiler and high pressure lines must be increased as the boiler pressure increases. Thermal efficiency increases as the boiler pressure is increased as shown in Figure 44. The metallurgical temperature limit for the turbine blades is approximately 1200°F. To obtain maximum heat transfer, the condenser should be operated at the highest possible temperatures. However, when



WORKING FLUID CONDITIONS AT:

(1) h = 160.4 BTU/#
T = 1196° F
P = 500 psia
S = .11316 BTU/# - °R
U = .177 ft/³#

(2) h = 121
T = 457.7
P = 1.0
S = .1379
U = 41.1

(3) h = 13.96
T = 457.7
P = 1.0
S = .02045
U = .00118

(4) h = 14.0
T = 457.7
P = 500
U = .00118

WHERE:

h = Enthalpy
T = Temperature
P = Pressure
S = Entropy
U = Specific Volume

FIG 42 SCHEMATIC, POWER PLANT

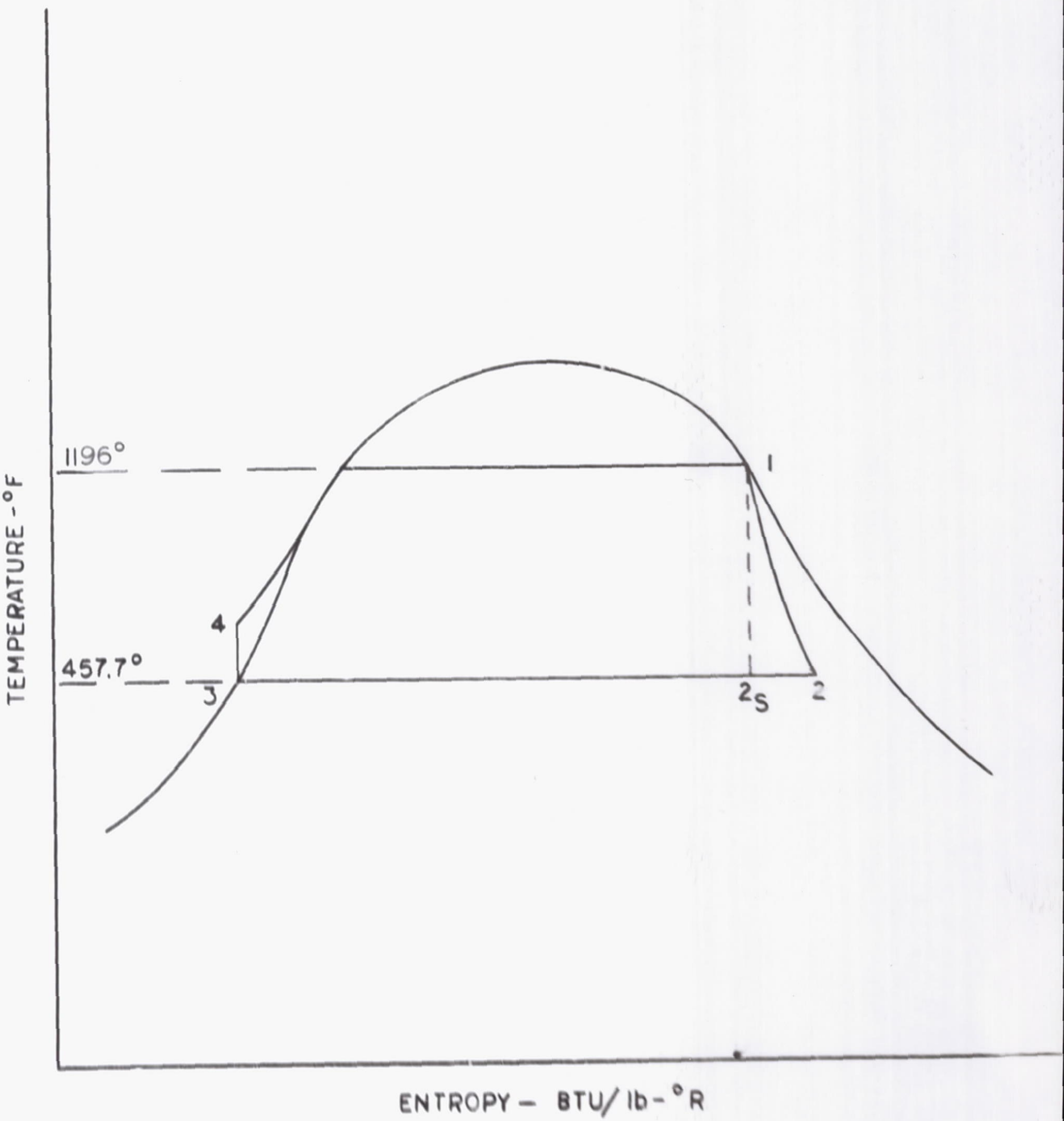
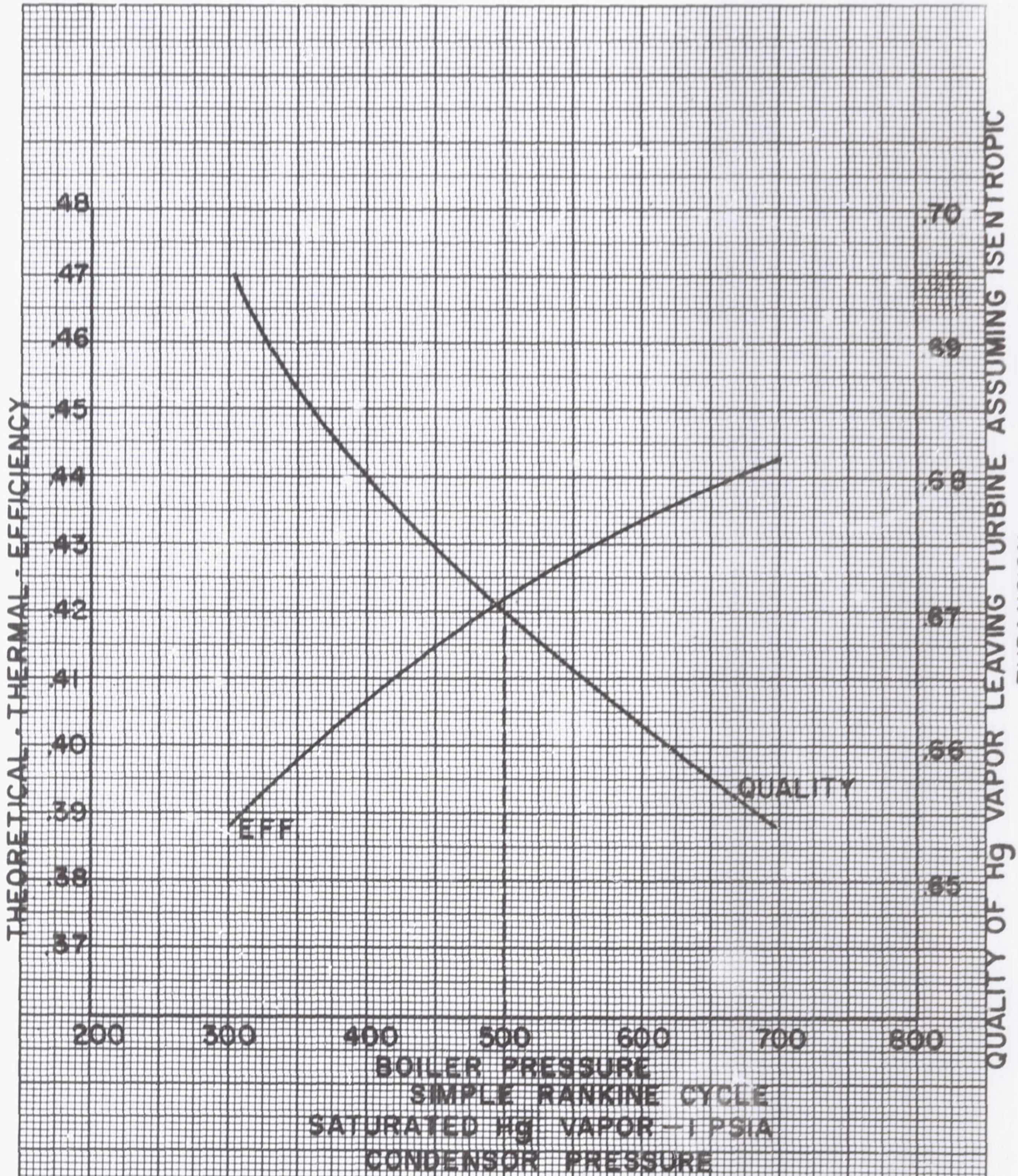


FIG.43 T-S DIAGRAM, RANKINE CYCLE



THEORETICAL THERMAL EFFICIENCY
QUALITY OF Hg VAPOR LEAVING TURBINE ASSUMING ISENTROPIC EXPANSION
.48
.47
.46
.45
.44
.43
.42
.41
.40
.39
.38
.37
.70
.69
.68
.67
.66
.65
EFF
QUALITY
200
300
400
500
600
700
800
BOILER PRESSURE
SIMPLE RANKINE CYCLE
SATURATED Hg VAPOR – 1 PSIA
CONDENSOR PRESSURE

FIG. 44

inlet conditions are constant, the thermal efficiency of the condenser decreases when the outlet temperature is increased. It is evident that a compromise must be reached in establishing optimum condenser operating temperatures. It has been determined that condenser area can be kept within reasonable limits and a satisfactory thermal efficiency obtained with the condenser operating at one psia and 457.7°F conditions. The overall efficiency of this power plant will be approximately 17.9% based on ratio of generator output to solar energy input. An energy flow diagram for this system is shown in Figure 45.

This system offers the following major advantages over other systems considered:

(1) Light weight - due to the fact that the fuel supply is solar energy, the components of the system are lightweight and the quantity of working medium (mercury vapor) is constant.

(2) The system is capable of continuous operation at maximum output of 975 watts throughout the daylight period. The vehicle may be halted for long periods to carry out scientific investigations without danger of consuming all the fuel and immobilizing the vehicle. During these periods the entire power output of the system is available for operation of instrumentation and the cooling system.

(3) Usable life of the vehicle is not limited by the power plant. With proper orientation of the parabolic reflector prior to darkness and proper preservation of instrumentation, the power plant is capable of operation during each succeeding period of sunlight.

b. Components (Figure 46)

(1) Boiler. The boiler for this power plant consists of the parabolic reflector, vapor lines, and evaporating vessel. The parabolic of revolution of the reflector will have a focal length of three feet and will be approximently seven and one-half feet in diameter. It will be made of lightweight sheet metal having up to 95% solar reflectivity. Geometric accuracy of the paraboloid must be very closely controlled. The evaporating vessel and vapor lines will be insulated with multi-layered metal foil with a very low infrared emissivity factor in order to minimize the heat loss.

(2) Turbine. It is generally true that the reciprocating engine is more efficient than the turbine for small power outputs. However, material compatibility problems are encountered when using mercury vapor as the working fluid. These material problems are more easily obviated in turbine design since there are less moving parts and there is more accumulated experience in solving these problems for turbine applications. Since the enthalpy drop of the vapor through the turbine is small, the velocity of the vapor leaving the nozzle will be small in comparison to the velocity in a steam cycle. Therefore, the mercury vapor turbine can operate with a lower blade speed and, consequently, lower rotational losses than a steam turbine. Based on preliminary calculations, a two-stage impulse turbine is recommended. Indications are that this turbine will have a maximum mechanical efficiency of approximately 50% and an operating speed of about 24,000 rpm. Speed will be controlled by a governor operating a by-pass valve which will route the vapor directly from the boiler to the condenser

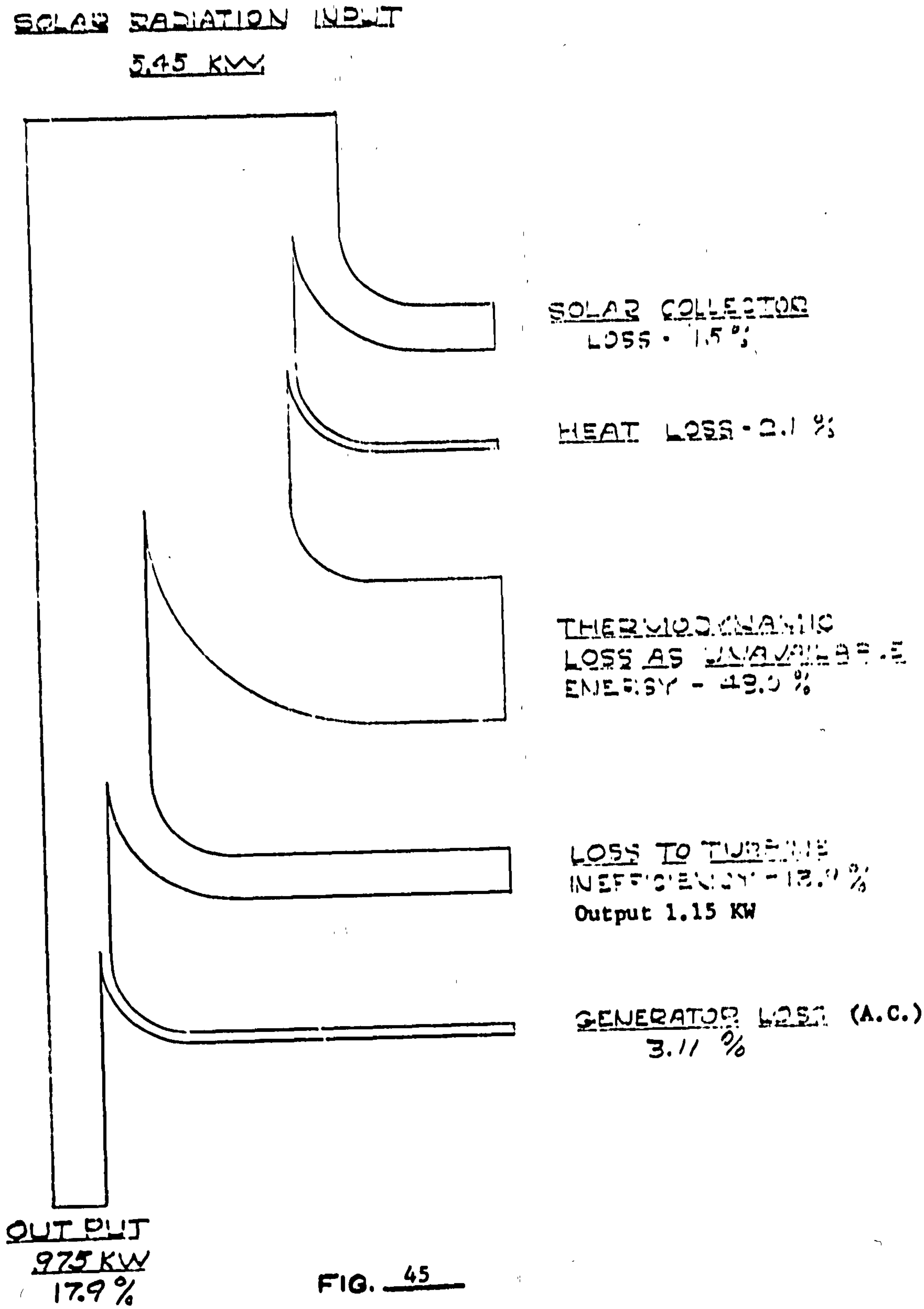


FIG. 45

ENERGY FLOW DIAGRAM FOR POWER PLANT

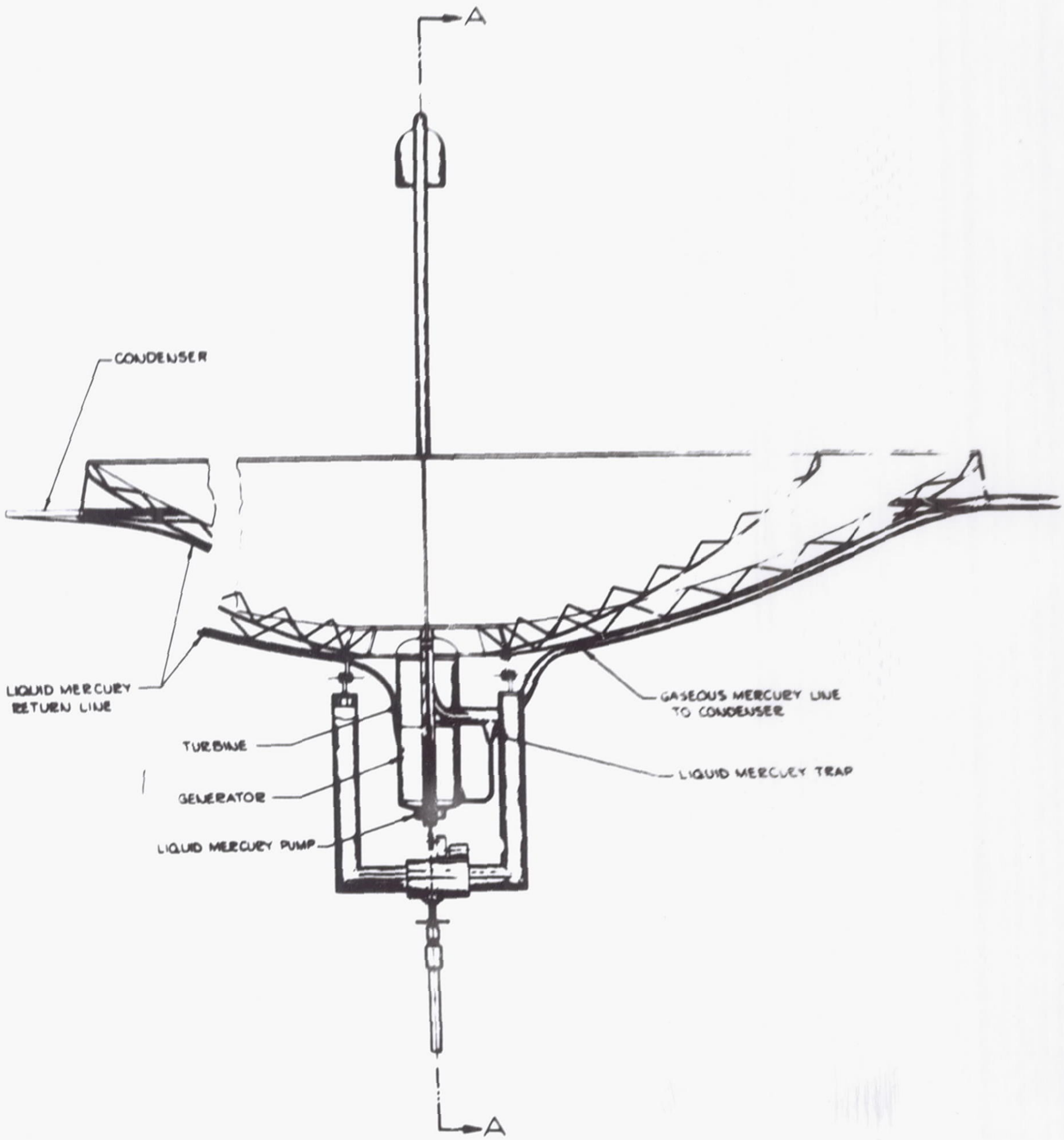
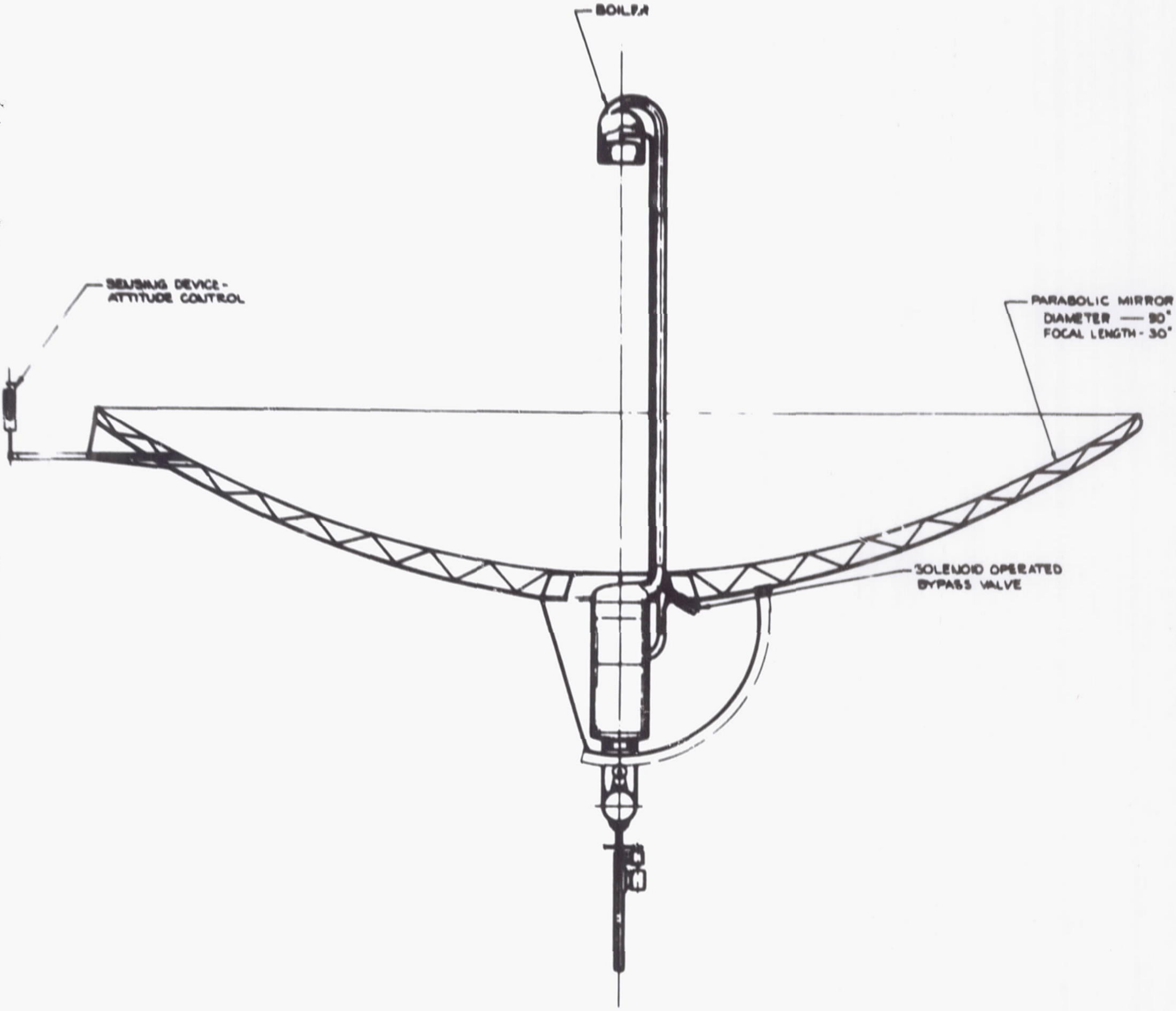


Figure 46A. MERCURY VAPOR POWER PLANT



SECTION A-A

Figure 46B. MERCURY VAPOR POWER PLANT

when required.

(3) Condenser. Due to the low operating pressure of one psia at 457.7°F the condenser may be made of lightweight stainless steel sheet metal. The outer surface of the condenser must be highly reflective to solar energy and highly emissive to infrared. Assuming an emissivity of 0.95, a radiating surface of 19.25 square feet will be required. The condenser design operating temperature of 457.7°F is well above the lunar surface temperature which makes it possible for the entire outer surface to be utilized for heat transfer and eliminates the requirement for constant reorientation as the vehicle moves about.

2. Alternate Power Plants

a. Open Cycle Turbo-Generator

(1) General. An open cycle turbo-generator power plant using hydrazine as the working fluid was investigated to determine the feasibility of utilizing such a power plant for the lunar roving vehicle. Due to the low thermal efficiency and high rotational turbine losses, the total propellant weight required for a 50 horsepower-hour power plant would be approximately 600 pounds. Since this greatly exceeds the 150 pounds total power plant weight (including propellants), no further consideration was given to this system.

(2) Characteristics.

From the relation $\frac{V_2^2 - V_1^2}{2 g J} = H_1 - H_2$

where $V_1 = 0$, $V_2 = 4200$ ft/sec * the change in enthalpy (Δ H) is found to be 352 BTU/lb.

* Altman, David and D. D. Thomas, Evaluation of Hydrazine as a Monopropellant and a Gas Generant, Progress Report No. 9-36, Jet Propulsion Laboratory, April 17, 1949.

The following energy losses were included in arriving at an overall turbine efficiency:

(1) Nozzle losses.

(2) Rotational losses due to friction and fanning action of idle blades in stages having partial admission.

(3) Leakage losses past the diaphragm shaft packing and past the blade tips.

(4) The leaving loss due to the kinetic energy at the exit of the last blade row.

(5) Loss of energy by conduction of heat through the turbine casing.

By adding the above losses, a turbine mechanical efficiency of 60 percent was estimated. The rotational losses represent the greatest turbine loss. This is due to the high rotational speeds of a two-stage, eight-inch diameter turbine or a three-stage, six-inch diameter turbine with a rotation speed of approximately 25,000 rpm. If it is assumed that the power requirement for the lunar power plant is 50 horsepower hours, the fuel requirement may be calculated as follows:

1 hp hr = 2544 BTU/hr

For 60 percent turbine efficiency the available change of enthalpy is 211.5 BTU/lb. (352 BTU/lb. x 60% efficiency)

Then, $\frac{2544 \text{ BTU/hr}}{211.5 \text{ BTU/lb}}$ = 12.0 lb/hr required

Hence, 50 hrs x 12.0 lbs/hr = 600 lbs of propellant required which is four times the total weight allowed for the power plant.

b. Hot Gas Motor Pump (Open Cycle)

(1) General. An investigation was also made on a piston type hot gas motor pump for lunar vehicle application. Even though this type power plant has a much higher thermal efficiency than the turbine, the total propellant weight required for 50 horsepower-hour operation would be approximately 220 pounds which exceeds the total power plant weight allowance. Therefore, no further consideration was given to this type power plant.

(2) Characteristics. Piston type hot gas motor pumps capable of delivering one to three horsepower and weighing approximately three pounds are commercially available, however, detailed specifications are not available at this time. Due to the unfavorable results obtained for the open cycle turbine system, calculations have been performed on the piston type using certain thermodynamic information from Figure 47 and the following relation:

$$\Delta S_{12} = C_p \, l_n \frac{T^2}{T_1} - R \, l_n \frac{P2}{P_1} \quad (3)$$

Where S_{12} = change in entropy from State 1 to State 2

C_p = specific heat (constant pressure)

R = Gas constant

P = Pressure

T = Temperature

By assuming no change in entropy and an exhaust pressure, the propellant necessary to drive the two horsepower motor pump can be calculated. For chamber temperatures around 1500°F the specific heat ratio (V) and the molecular weight (M) can be found from Figure 47. Using these

values and an assumed exhaust pressure of 1.0 psi in equation (3) above, a change in enthalpy of 970 BTU/lb is determined. If a mechanical efficiency of 60 percent is assumed, the total propellant weight required is approximately 219 pounds which exceeds the total power plant weight and eliminates this system from further consideration.

c. Solar Cells. Since solar radiation on the lunar surface has such a high intensity, the use of solar cells as a means of power conversion was investigated. Solar cells have the characteristics of losing efficiency as their operating temperature increases. Under ideal laboratory conditions, solar cells will operate at an efficiency as high as 10-12%. However, for use on the lunar roving vehicle, the conditions are anything but ideal. An operating temperature of around 200°F can be expected with a resulting efficiency of from 4 to 5%.

Vehicle power requirements, including both mobility and electronics equipment, amount to approximately 650 watts. Solar cell area required for this power output would be about 160 square feet. It is obvious that with this much area required the use of a solar cell power plant would result in a very impractical vehicle.

d. Hydrogen-Peroxide. The most efficient power plant using H_2O_2 as a monopropellant would require about 8 lbs of fuel per hp-hr. Again assuming a 50 hp-hr power requirement, the required fuel would weigh 400 lbs which exceeds the 150 lb weight limitation.

e. Hydrogen-Oxygen. From an efficiency and weight standpoint the hydrogen-oxygen power plant is the best as far as open cycle systems are concerned. It appears from published data and preliminary calculations that 2.5 lb/hp-hr can be expected. Again assuming a 50 hp-hr

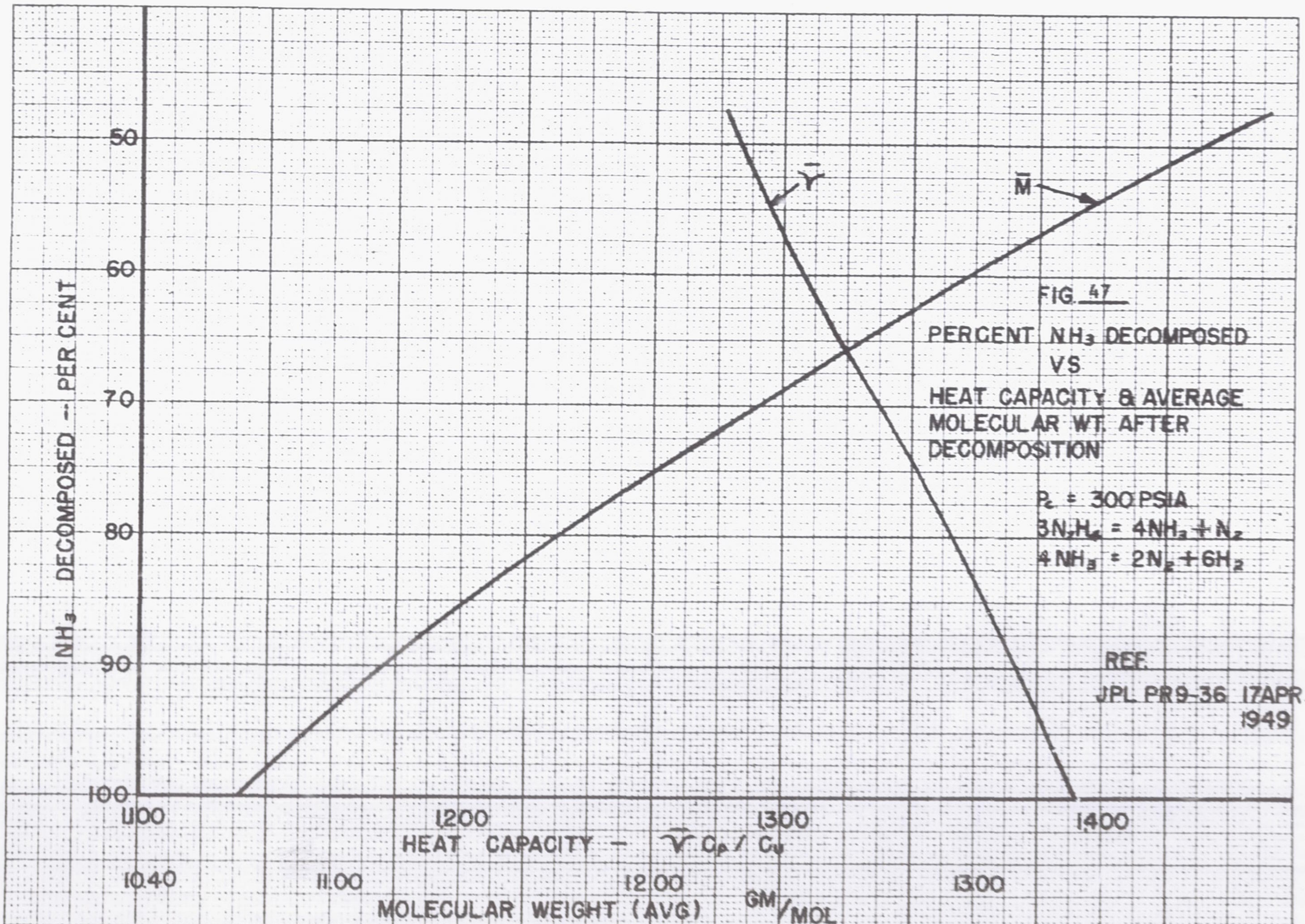


FIG. 47
PERCENT NH3 DECOMPOSED
VS
HEAT CAPACITY & AVERAGE
MOLECULAR WT. AFTER
DECOMPOSITION
Pc = 300 PSIA
3N2H4 = 4NH3 + N2
4NH3 = 2N2 + 6H2
REF.
JPL PR9-36 17APR 1949
M̄
γ̄
50
60
70
80
90
100
NH3 DECOMPOSED -- PER CENT
1100
1200
1300
1400
HEAT CAPACITY — γ̄ Cp / Cv
10.40
11.00
12.00
13.00
MOLECULAR WEIGHT (AVG) GM/MOL

requirement, the fuel requirement would be 125 lbs. Cryogenic storage of the Hydrogen and Oxygen for 14 earth days under lunar environmental conditions would be a difficult problem. Additional fuel would have to be provided to compensate for losses due to boil off. Preliminary investigations indicate that this boil off rate could not be kept low enough to make this system practical.

F. ELECTRICAL POWER SYSTEM

1. General

Determination of the electrical power system to be used on the lunar roving vehicle is dictated by certain known requirements and characteristics necessary for operation on the lunar surface and by electrical equipment which will be available by the proposed date of use. Two basic systems, A. C. and D. C., were studied and an optimum solution for this application was extrapolated from calculations and equipment data. The selection of the electrical generating and drive motors is governed by the following general requirements:

a. The mechanical driving unit should operate at a very high rpm (in the order of 25,000 rpm) with a maximum continuous power output.

b. Operate in a vacuum.

c. Operate at temperatures in the order of 600°F.

d. Weight should be kept to a minimum.

e. Efficiency should be high.

f. Operate for long periods of time without external control.

In addition to the above general requirements, a small storage battery and conversion pack must be included in the system to permit approximately 10 minutes emergency operation of the vehicle. This will enable the vehicle to be withdrawn from a shadow in the event time delay in signals permits it to travel into a shadow that obsecures the solar mirror.

2. Proposed System (A.C.)

a. Generator. A one KVA, three-phase, 400 cycle generator will meet the necessary power output requirements. With a permanent magnet

type construction, it will operate without sliding or exposed electrical contacts. No problems are anticipated when operating at the required high ambient temperatures and in a vacuum.

b. Motors. One presently available A. C. motor which meets these general requirements is the Hypersyn motor. It is a multiple pole motor which combines the desirable features of the induction, hysteresis, and the D. C. excited synchronous motor. It is available in sizes from 1/100 to one horsepower for any frequency, voltage, and speed, and will operate in a wide range of ambient conditions such as under water, in a vacuum, etc. It can be made in special shapes to fit a variety of applications. It embodies a new rotor design which can be operated single or polyphase. Essentially it is a solid magnetic structure with four regions having different electrical and magnetic properties. The rotor is of a permanent magnet type and no sliding contacts are required on the rotating portion of the motor. Motor design indicates that an efficiency of approximately 85% is obtainable, that it can be operated at sychronous shaft speeds as low as 300 rpm and at lower shaft speeds as a reasonably efficient induction motor. Present data indicates that a 0.25 hp motor can be housed in a case 2" long and 4½" in diameter and will weigh six pounds. Two of these motors will be required for the drive system.

c. System Characteristics. Based on the general requirements stated above the following characteristics of the A. C. system have been determined:

(1) 24,000 rpm is satisfactory for an A. C. alternator.

(2) The A.C. units have no sliding or exposed contacts and

can be designed to operate satisfactorily in a vacuum at temperatures of 600°F.

(3) Estimated weight of the A.C. electrical system is 25 pounds. When added to the estimated power plant weight of 110 pounds, this will give a total power system weight of 135 pounds which is within the allowable weight.

(4) The overall system efficiency is satisfactory as shown by the following calculations based on known A.C. component characteristics.

<u>Alternator Output</u>

Input to alternator (From Figure 45)------------1150 watts

Alternator output-------------------------------- 975 watts
(Efficiency 85% @ 24,000 rpm).

<u>Drive Power Requirements</u>

Design vehicle drive load is 0.38 hp (283 watts) at 3 miles/hr on a 15° slope requiring 380 foot pounds torque (From Figure 38).

Drive Shaft Output @ 5¼ rpm and 400 ft. lb.----- 283 watts

Required Input to Reduction Gears--------------- 349 watts
(Efficiency 81% @ 300 rpm in - 5¼ rpm out)

Required Input to Motor (Total Drive Load)------ <u>411 watts</u>
(Efficiency 85% @ Rated RPM)

Where conditions other than design conditions are encountered, such as crossing a boulder or other obstacle requiring higher torque, the speed will be reduced and drive shaft power available will also be reduced due to lower motor operating efficiency as shown below. However, the total drive motor load will remain the same as shown below.

Drive Shaft Output @ 1 rpm and 1200 ft. lb.---- 162 watts

Required Input to Reduction Gears-------------- 200 watts
(Efficiency 81% @ 57 rpm in - 1 rpm out)

Required Input to Motor - Total Drive Load----- 411 watts
(Efficiency 49% @ 19% rated rpm)

Cooling System Power Requirements

The net instrument compartment cooling load will be 0.469 hp (350 watts) (from cooling system calculations). Therefore the total input to the compressor drive motor will be:

Shaft Output - Compressor Drive Motor---------- 350 watts

Required Input to Motor (85% eff.)-------------- 411 watts

The total continuous power load for instrument operation is assumed to be 100 watts and the vehicle control load is estimated to be 14 watts. However, the total power output of 975 watts less the cooling load will be available for special instrument operation such as sample collector operation, when the vehicle is halted. The total power requirements will be:

Drive Load-------------------------------------	411 watts
Cooling Load-----------------------------------	411
Instrument Load--------------------------------	100
Control Load-----------------------------------	14
Total Load------------------------	936 watts

Since the generator output will be 975 watts the net power available for charging a battery pack and/or other contingencies will be:

Generator Output-------------------------------	975 watts
Calculated Load--------------------------------	936
Power Available for Other Uses	39 watts

3. Alternate Power System (D.C.)

Investigation of a D.C. power system based on known components was made and it was determined that the system was unsatisfactory due to the following characteristics.

a. 12,000 rpm is the nominal limit for a D.C. generator if excessive brush wear is to be avoided. This is unsatisfactory for direct turbine drive.

b. The standard D.C. system requires the use of brushes. The effects of lunar environment on exposed brush operation have not been determined. However, it is felt that many technical problems would be encountered. The requirement to operate at 600°F temperature will require special brush holder construction and special brushes. Special seals would be required if pressurized containers were required. Brushless D.C. motors have been built, but only in small fractional horsepower sizes. This technique requires considerable development and operating efficiencies are not known at this time.

c. Total weights for D.C. systems were estimated based on full time and part time operations as shown below. In all but one case of intermittent operation the estimated weights exceeded the allowable weight. These calculations were made by taking the total wattage load calculated, rating this by percent of available capacity, and estimating the weight of the mechanical unit from this percentage.

	Elec. System Weight (Lbs).	Power Plant Weight (Lbs).	Total Weight (Lbs).
<u>Full Time Operation</u>			
Rated Speed - High Torque	40	215	255
<u>17% Operating Time</u>			
1 hr in 6 hrs	80	80	160
10 min. in 60 min.	42	82	124

On the part time operation system, batteries would be used to supply peak load demands, thus minimizing power plant output requirements and size of plant.

d. The overall efficiency of a D.C. system would be unsatisfactory as shown by the following calculations:

<u>Generator Output</u>

Input to Reduction Gear (From Figure 45)---------1150 watts

Input to Generator-------------------------------1035 watts
(Efficiency 90% @ 12,000 rpm out)

Generator Output--------------------------------- <u>828 watts</u>
(Efficiency 80%)

<u>Drive Power Requirements</u>

Design Drive Load 0.38 hp - 283 watts (Figure 38)

Drive Shaft Output @ 5¼ rpm and 400 ft. lb.------ 283 watts

Required Input to Reduction Gear----------------- 393 watts
(Efficiency 72% @ 4000 rpm in - 5¼ rpm out)

Required Input to Motor (Total Drive Load)------- <u>562 watts</u>
(Efficiency 70% @ rated rpm)

For unusual conditions such as crossing a boulder or climbing a hill requiring higher than design torque, power requirement will

increase even though the speed will still decrease.

Drive Shaft Output @ 2 5/8 rpm - 1200 ft. lb.----- 424 watts

Required Input to Reduction Gears------------------ 590 watts
(Efficiency 28% @ 2000 rpm in)

Required Input to Motor---------------------------1124 watts
(Efficiency 57% @ 1/2 rated rpm)

Cooling System Power Requirements

Assuming the same cooling system load of 350 watts as before:

Shaft Output-Compressor Drive Motor--------------- 350 watts

Required Input to Motor (70% eff.)---------------- 500 watts

Thus, assuming a maximum continuous D.C. input of 105 watts to the instrument compartment and 14 watts for controls the total load for normal operating conditions will be:

Drive Load @ rated rpm	562 watts
Cooling Load (D.C. Drive)	500
Electronic Equipment (D.C. Input)	105
Control Load	14
Emergency Battery Pack	39
Total Load	1220 watts

When operating under other than normal conditions the load will be:

Drive Load @ 1/2 rated rpm	1124 watts
Cooling Load (D.C. Drive)	500
Electronic Equipment (D.C. Input)	105
Control Load	14
Emergency Battery Pack	39
Total Load	1782 watts

Obviously, either of the above cases requires far more power than the 828 watts output of the generator. Therefore, a full time operation D.C. system is considered unsatisfactory for use with the proposed power plant.

G. LUNAR LANDING SHOCK ABSORBER

1. General Design Parameters

Present planning for the final phase of the landing sequence for the lunar payload (stationary package and roving vehicle) is a free fall to impact following a braking stage to achieve a full stop. The permissible impact velocity is somewhat higher than the planned velocity to be attained during free fall to allow for any residual velocity which might remain after the last braking stage. Established parameters for this phase of the landing which affect the shock absorber design are:

a. Maximum permissible impact velocity - 17-20 M/Sec (60 ft/sec mean)

b. Maximum permissible deceleration - 20 g (Earth)

c. Gross payload weight - 2175 lbs (Earth) 362 lbs (Lunar)

d. Maximum weight of shock absorber and mounting - 275 lbs

e. Same shock absorber design to be used for both the stationary package and roving vehicle.

f. Contamination of lunar surface at point of landing is permissible for the roving vehicle but not for the stationary package.

g. Payload will have attitude control during free fall phase to insure proper landing attitude.

h. Maximum permissible take off diameter of shock absorber device is 120 inches (missile payload diameter).

i. Rebound of the payload after impact must be minimized.

j. Stationary package must come to rest with unrestricted access to lunar surface for core drill and sesimograph operations.

k. Length of shock absorber device must be held to minimum.

2. Basic Calculations

The kinetic energy stored in the payload upon impact with the lunar surface will be:

$$E = 1/2\ MV^2 = \frac{2175 \times 60^2}{2 \times 32.2} = 121{,}000 \text{ ft lbs (154 BTU)}$$

Since the maximum permissible deceleration is 20g (earth), the maximum force of deceleration will be (assuming a rigid lunar surface):

$$F = Ma = \frac{W}{g} \times 20g = 20w = 20 \times 2175 = 43{,}500 \text{ lbs}$$

To prevent rebound, all of the kinetic energy must be absorbed during impact. Therefore, the required deflection (S) of the shock absorber device must be:

$$S = \frac{E}{F} = \frac{121{,}000 \text{ ft lbs}}{43{,}500 \text{ lbs}} = 2.8 \text{ ft.} = 34 \text{ in.}$$

Time of deceleration will be:

$$t = \frac{2\ S}{V} = \frac{2 \times 2.8}{60} = 0.093 \text{ Sec.}$$

3. Discussion

An ideal solution to the deceleration problem for the lunar payload would require the shock absorber to absorb the 121,000 ft. lbs. of energy in a distance of 2.8 feet with a constant deceleration force of 43,500 pounds. Under such ideal conditions the load deflection curve would appear as shown in Figure 48A. With constantly increasing deceleration force from zero to maximum allowable the curve would appear as shown in Figure 48B. However, the load-delfection (stress-strain) curves of common materials do not coincide with either of these conditions. The curves for most common structural materials will appear somewhat as shown in Figure 49A. The curves will vary for each material depending upon its proportional limit, yield strength, ductility, ultimate strength, etc.

The stress-strain curve of some industrial type shock absorbing materials and metals may depart from the straight line portion of stress-strain curves (Hooke's Law) as shown in Figure 49B. However, unless the rupture strength of the material is exceeded the elastic hysteresis will cause the material to partially return to its original condition as shown by the dotted line in Figure 49B. This is desirable for industrial applications but would cause undesirable rebound in the lunar payload application. Other undesirable features of such materials are high unit weights and small unit strain (deflection) which would require excessive length and weight for the payload shock absorber.

Other major considerations in the shock absorber design are the effects of other than vertical impact, lateral motion of payload at impact, and impact on uneven terrain.

A shock absorber for this application should be relatively simple in operation and at the same time it must have the necessary characteristics to insure complete reliability under somewhat unpredictable impact circumstances. Very accurate calibration will also be most important. It is not intended to propose a solution to this problem at this time. Several possible solutions are discussed below to indicate the range of possibilities being considered. Each possibility listed, as well as others that may develop, must be more thoroughly investigated and tested before arriving at a "best" solution.

4. Possible Solutions

a. Corrugated Cardboard. A common crushable material considered for use in the shock absorber was corrugated paper. A stress-strain curve obtained by static loading a sample of this material is shown in

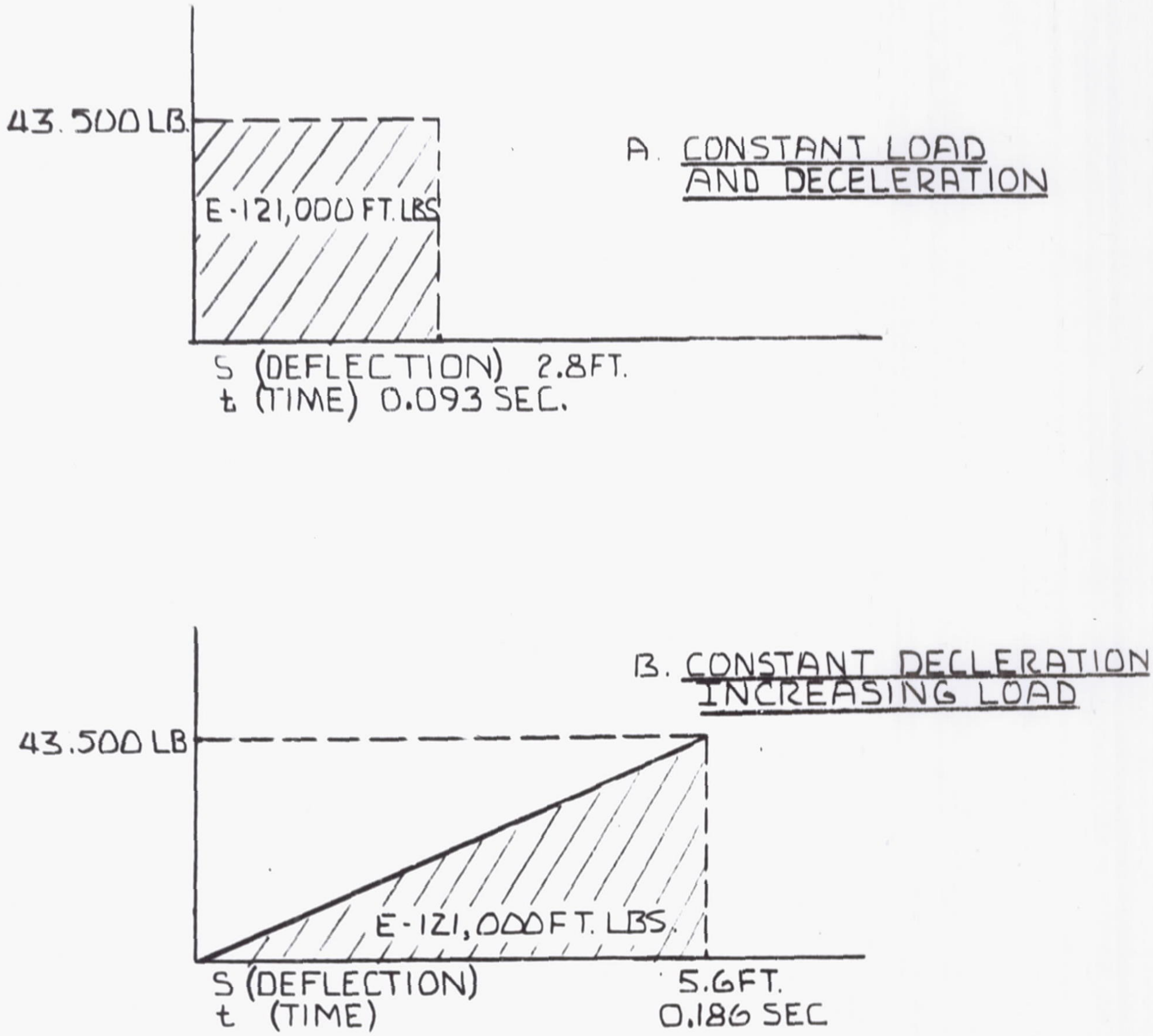
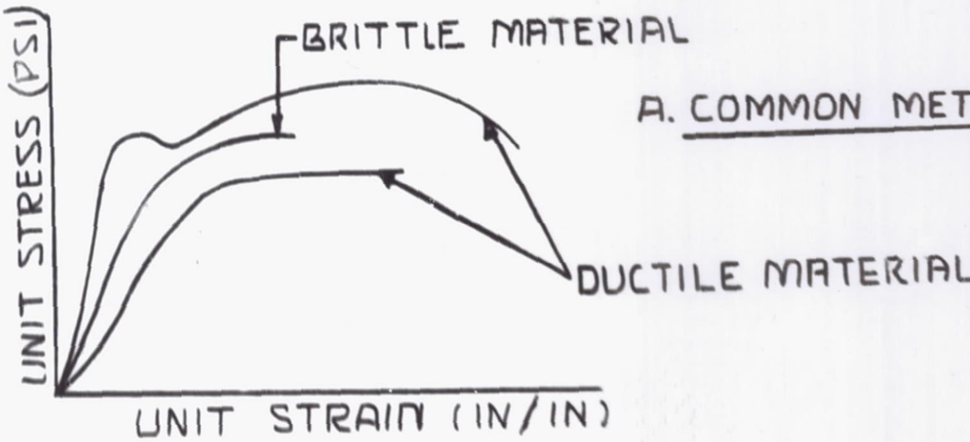
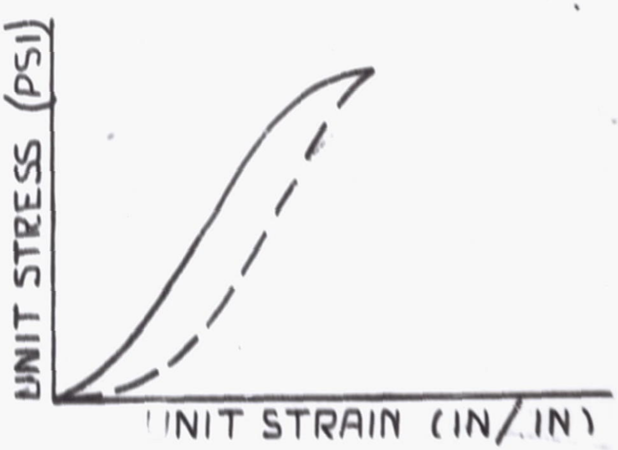


FIG. 4.8 THEORETICAL LOAD DECLERATION CURVES



A. COMMON METALS



B. INDUSTRIAL SHOCK ABSORBER

FIG. 49 TYPICAL STRESS - STRAIN CURVES

Figure 50. It shows that a rather constant load is held with increasing strain as desired. However, total strain is only about 50% of original length which would require an initial shock absorber length of approximately six feet. Calculations show that approximately 570 pounds of material would be required to absorb the 121,000 foot pounds of energy in three feet of deflection. The acceleration time history of this material under impact loading is not known, but is not expected to be favorable.

b. Cellular Styrofoam. A stress-stain curve obtained from a static loading test of a common cellular plastic styrofoam is shown in Figure 51. A relatively large amount of energy is absorbed under constant load which is desirable. However, the total strain (useable deflection) is only about 50% of the initial length of the sample tested. Calculations show that only 26 pounds of this material would be required to absorb the landing shock of the payload. However, this material has other characteristics as listed below which preclude its use as a lunar shock absorber.

(1) Becomes brittle at impact load rates resulting in an unsatisfactory deceleration time history.

(2) Life in a vacuum is unknown but doubtful.

(3) Cells contain gas that would be released on impact.

c. Wires and Diaphragms.

(1) General. The mechanical properties of copper and aluminum are well known at impact loading rates and these properties are not expected to be adversely affected by vacuum conditions. The

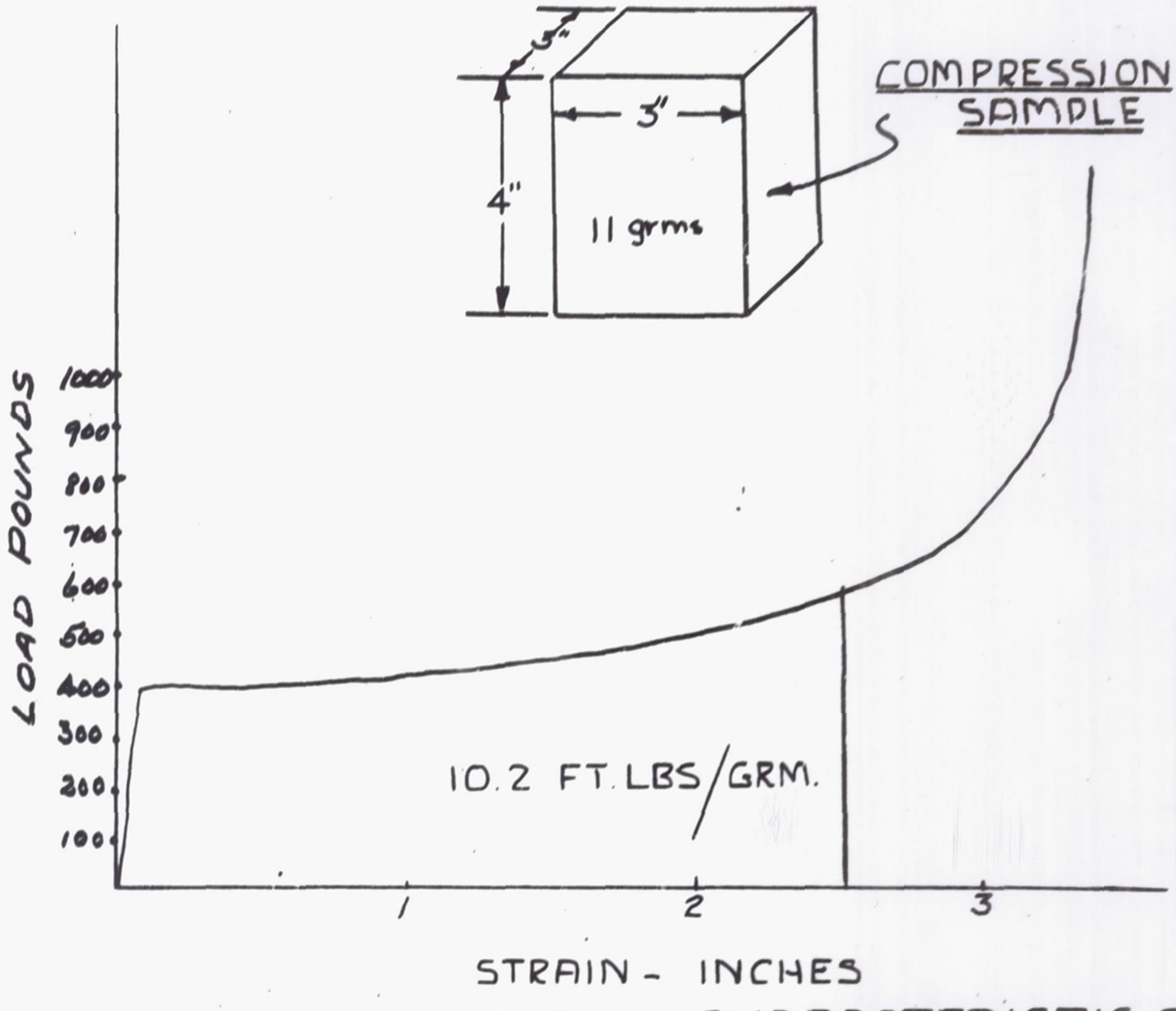


FIG. 2-51 LOAD-STRAIN CHARACTERISTIC OF STYROFOAM (STATIC LOADING)

amount of energy absorbed by these metals upon deformation increases with increasing loading rates and decreasing temperatures. Since the temperature within the shock absorber can be controlled by controlling the reflectivity of the outer surface, almost ideal energy absorbing conditions could be established.

(2) <u>Copper Wires</u>. A stress-strain diagram of a soft copper wire under static loading is shown in Figure 52. Calculations show that approximately 40 pounds of wire would be required to absorb the energy load at impact. The total strain exhibited is only about 35% of original length. However, by arranging various lengths of coiled wires in a device similar to that shown in Figure 53 the total strain of all wires could be utilized. It would be necessary to carefully select the length and size of the wires to obtain an overall stress-strain curve which would show a constant deceleration load over the minimum deceleration distance. However, the total required length of the device would be in excess of six feet or twice the minimum deceleration distance.

(3) <u>Aluminum Diaphragms</u>. An alternate method of utilizing the energy absorbing properties of deforming metal is shown in Figure 54. This method would probably be easier to "package" than the wire scheme described above but would still require about the same total length of device. In this scheme, a stack of thin aluminum sheets would be arranged within a round container. Upon impact the smooth rounded plunger would deflect and finally rupture successive sheets or diaphragms. Several sheets would be in contact at any one time. The shape of the plunger would insure that maximum area of each sheet was

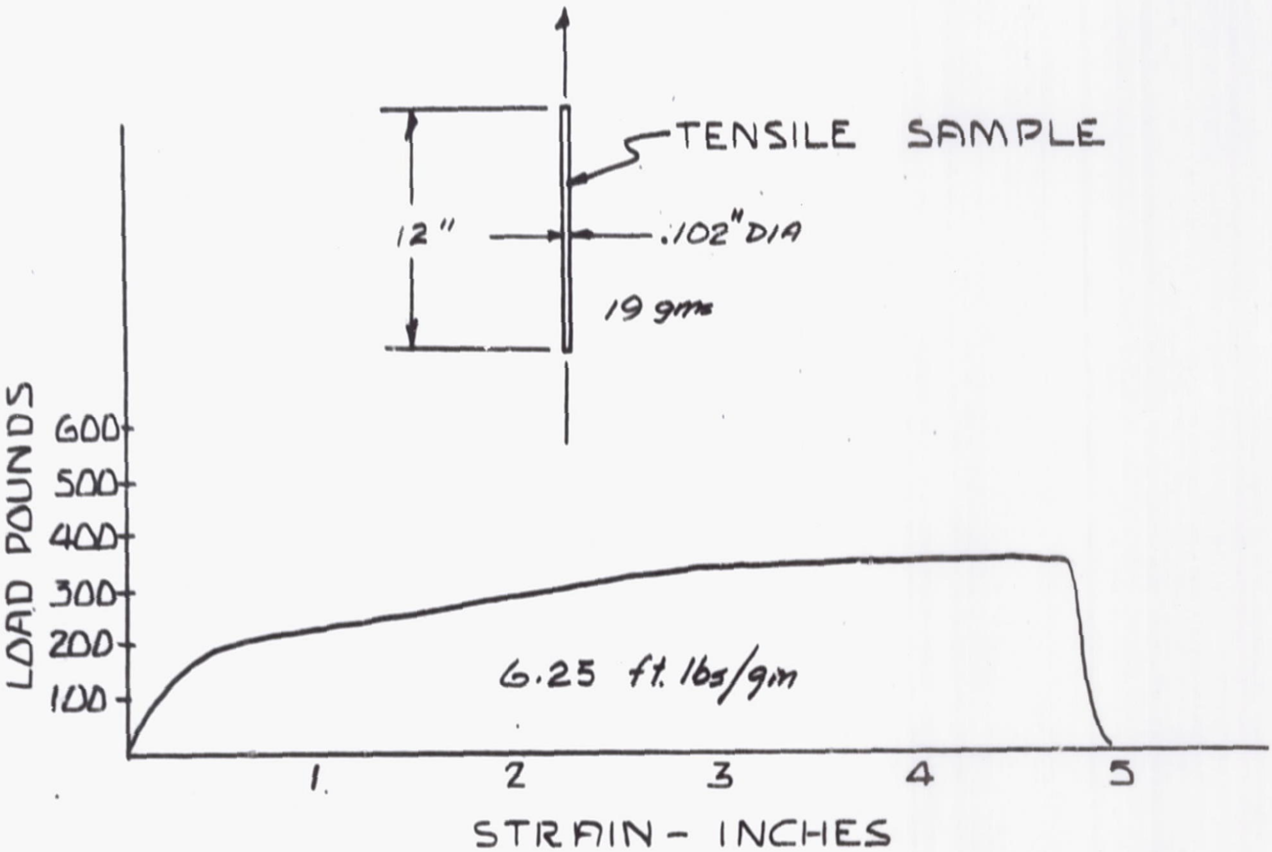


FIG. 3-52 LOAD-STRAIN CHARACTERISTIC OF SOFT COPPER WIRE (STATIC LOADING)

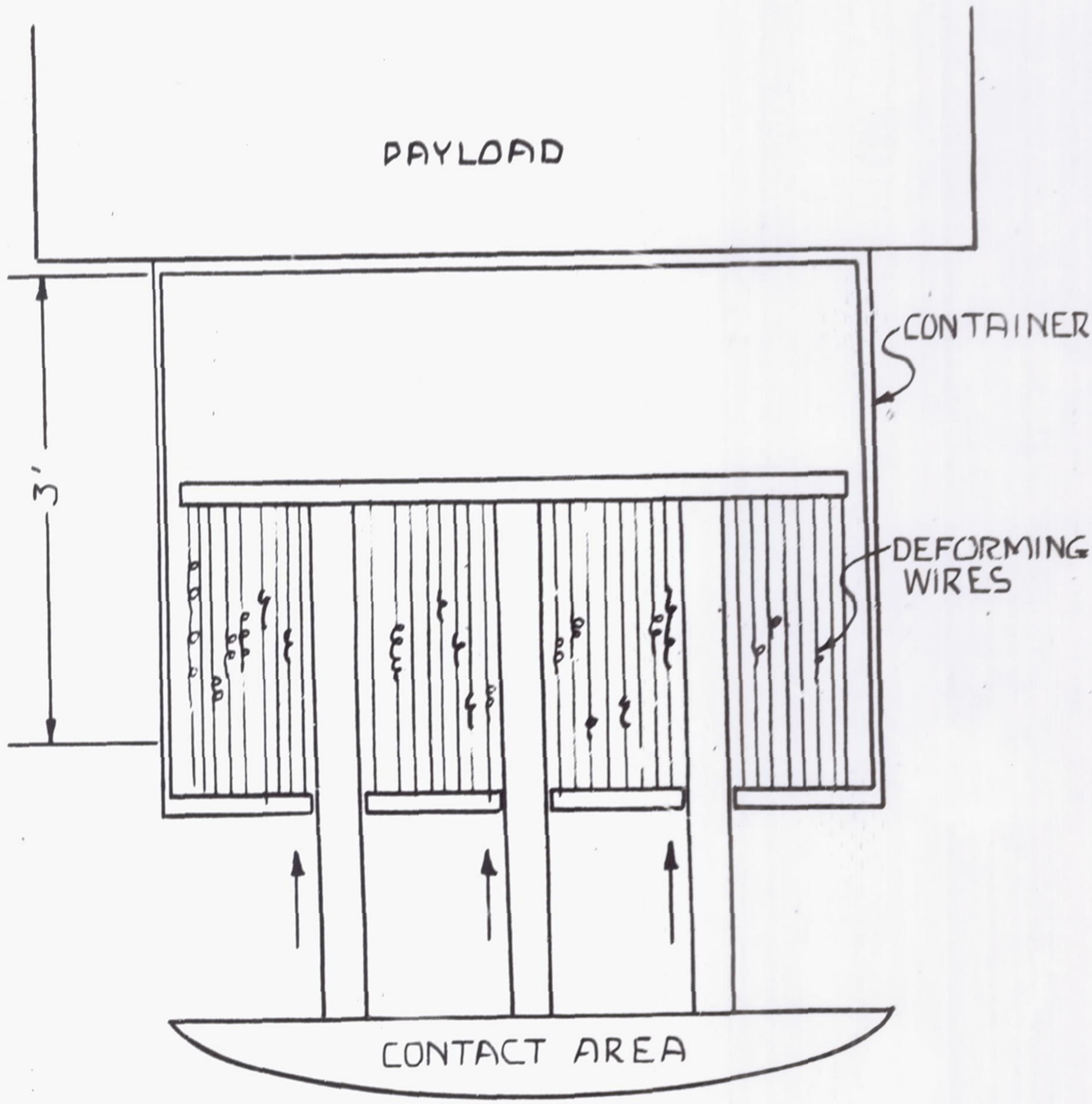


FIG. 53 POSSIBLE ARRANGEMENT FOR ABSORBING SHOCK BY TENSILE DEFORMATION OF WIRES

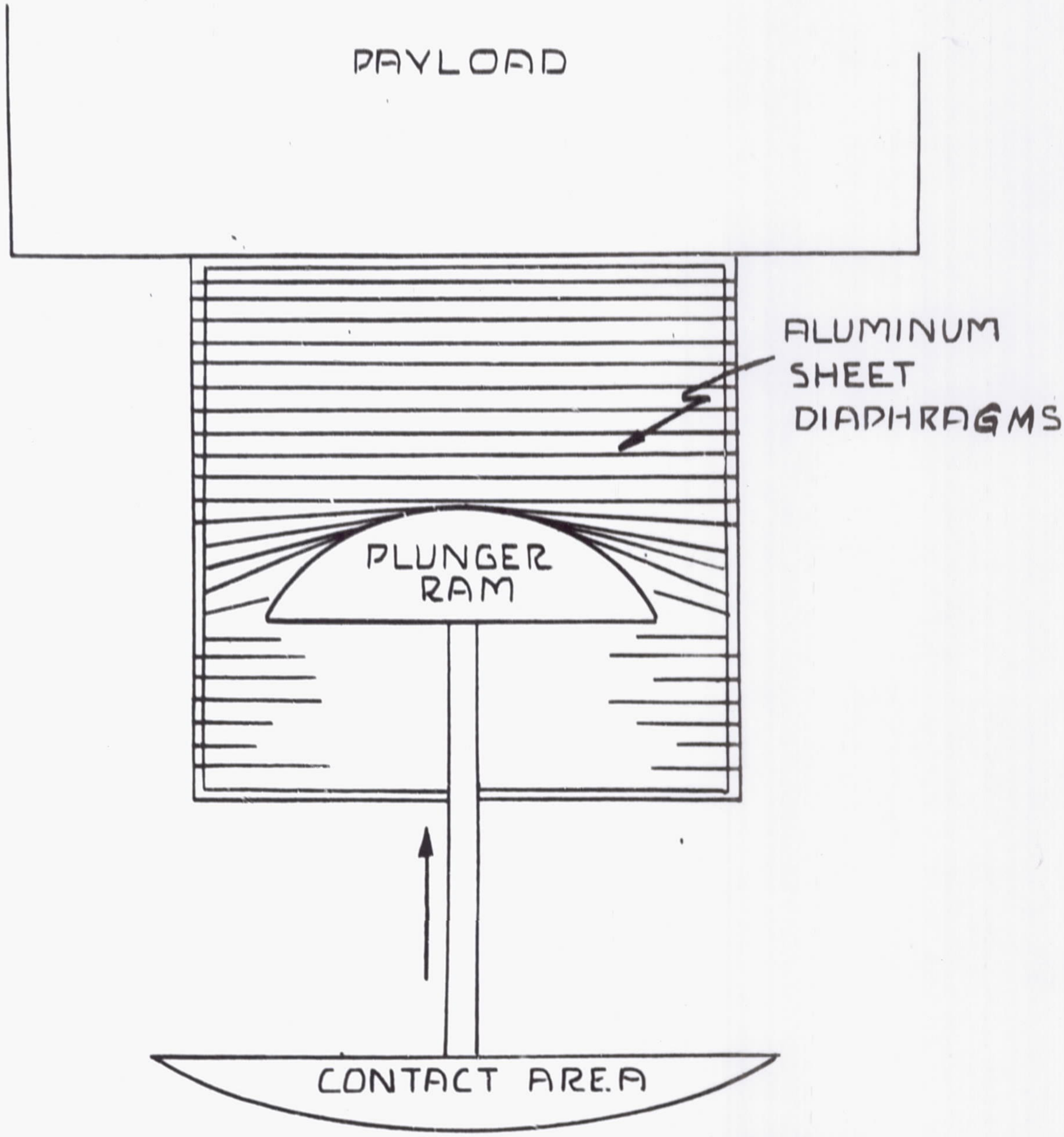


FIG. 6-54 ARRANGEMENT FOR ABSORBING SHOCK BY DEFORMATION AND RUPTURE OF MEMBRANES

96

plastically deformed to the fullest extent before rupture, thereby reducing the total weight of material required.

d. Shearing and Deforming Metal. Two possible methods of combining shear and deformation of metal to absorb the landing shock are illustrated in Figures 55 and 56. Both devices illustrated utilize cutting knives on the lower or contact plate to shear a cylindrical sleeve along vertical milled lines as the sleeve is forced against the cutting edges by the force of impact. In order to utilize the maximum deformation properties of the metal, the sheared strips of metal are forced into concentric coils by the shape of the cutting knife footing. The same principle is employed in both devices illustrated. However, the single column device shown in Figure 56 has the added feature of a ball-joint mounted contact plate to permit landing at an angle or on uneven surfaces. Both devices have the advantage of minimum length requirements. However, as with all other schemes mentioned, the problems of lateral motion during impact and other than vertical impact have not been completely solved.

e. Gas Bag. Although it has been stated that the use of a gas filled bag as a shock absorber is undesirable (for stationary payload) due to contamination of the surface by venting or out-gassing, the characteristics of one such scheme are presented for further consideration. This scheme would employ a gas filled bag to absorb the energy at impact by compression and throttling of the gas. The vented gas would not be allowed to escape to the surrounding space but would be contained by a second and larger bag as illustrated in Figure 57. This scheme would offer the advantages of maximum deceleration force through

minimum distance resulting in minimum length of shock absorber. Vibration would also be reduced to a minimum by this method and it is felt that overall weight would be reduced due to the fact that a gas would be used to absorb the energy. This system would also partially alleviate the problem of other than vertical landing and lateral motion during impact. Illustrative calculations are shown below based on the assumption that Hydrogen gas will be used and that the shock absorber bag will be 10 feet in diameter, 2.8 feet long, and initial pressure will be 3.8 psia at $0^{\circ}F$. Volume of the bag will be 220 cubic feet.

Where:

M = Total weight (lbs.)

P = Pressure (lbs/in^2)

V = Volume (ft^3)

R = Constant (ft lbs/lb $^{\circ}F$)

T = Temperature ($^{\circ}F$)

E = Energy (BTU)

C_v = Specific Heat at Constant Volume (BTU/lb $^{\circ}F$)

Then:

$$M = \frac{PV}{RT} = \frac{(3.8)\ (144)\ (220)}{(767)\ (460)} = .34 \text{ lb.}$$

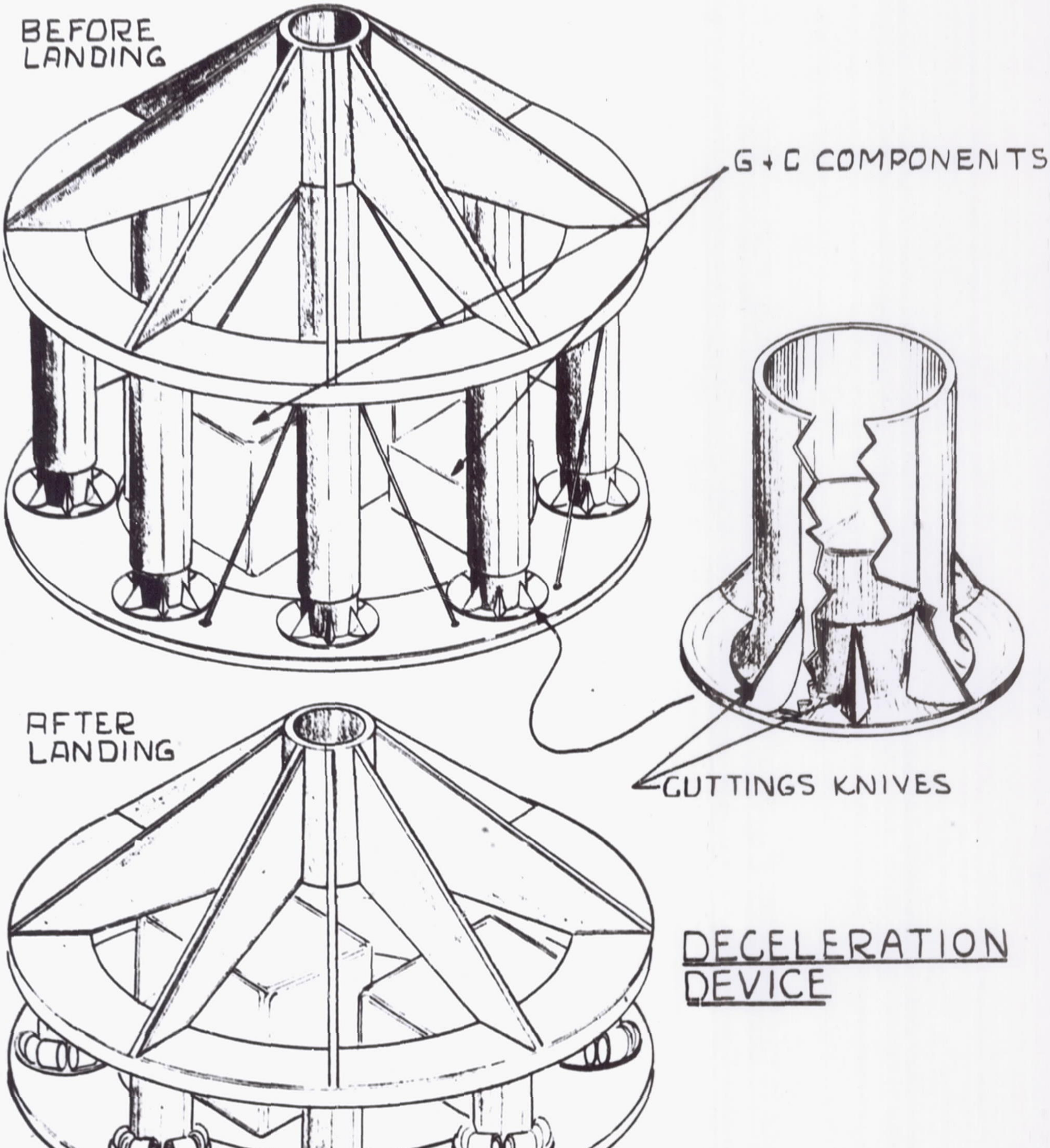
$$\Delta E = M C_v \Delta t$$

$$154 = (.34)\ (2.43)\ (T_2 - T_1)$$

$$T_2 = \frac{(154)}{(.34)\ (2.43)} + 460 = 646^{\circ}R = 186^{\circ}F$$

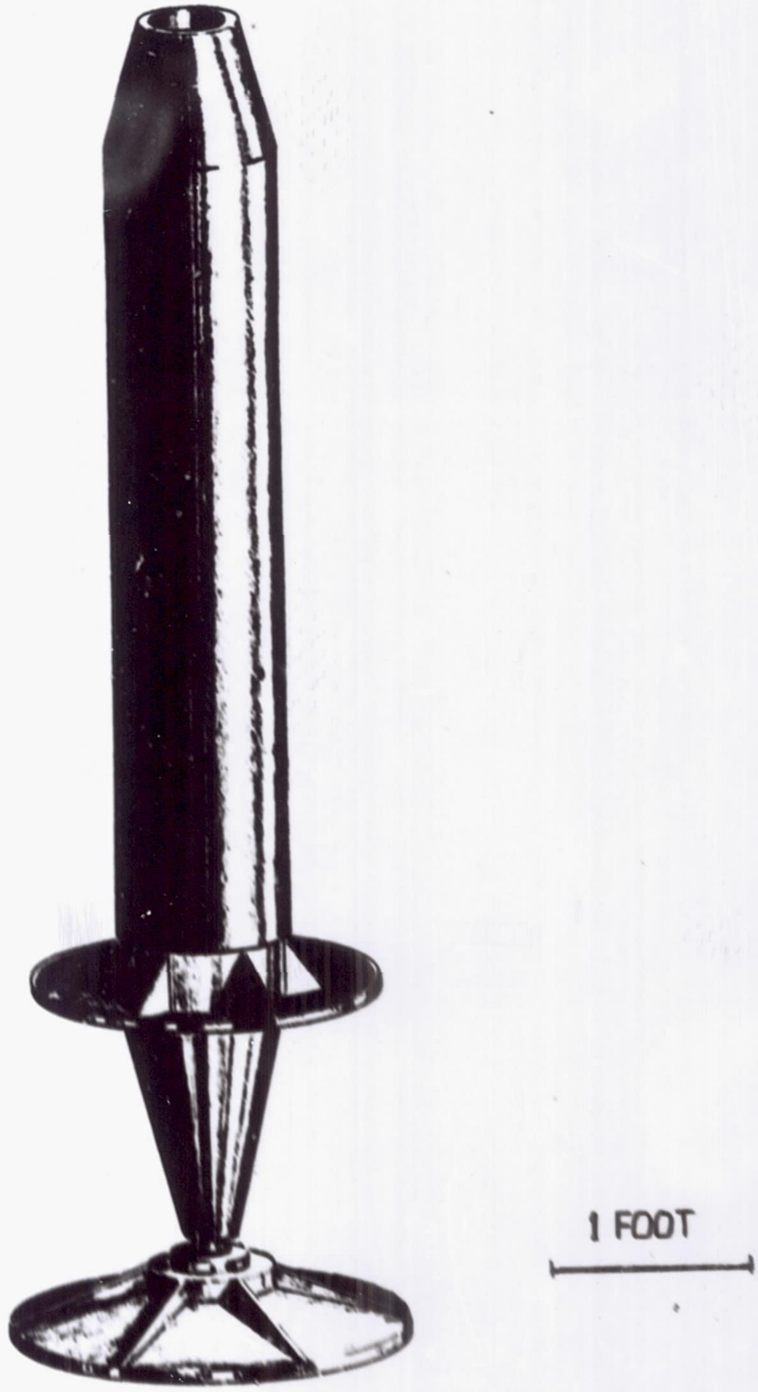
$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

$$V_2 = \frac{(220)\ (646)}{460} = 310 \text{ ft}^3$$



DECELERATION DEVICE

FIG. - 55



DECELERATION DEVICE

FIG. 56

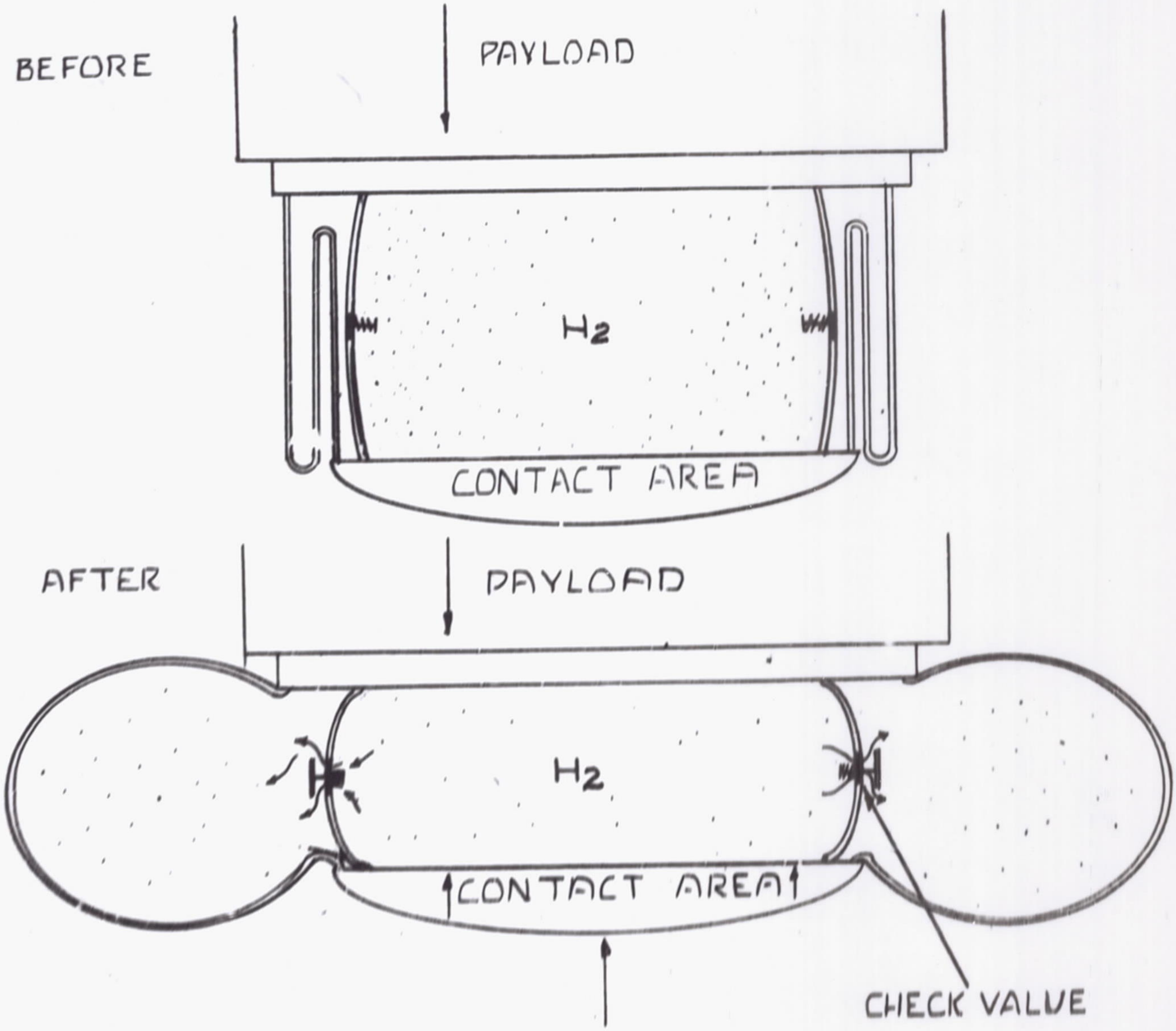


FIG. 57. GAS BAG - DECELERATION DEVICE

5. Conclusions

Obviously there are many other materials and combinations of materials which must be investigated before arriving at a solution to this problem. Much of the work previously accomplished in this area was directed toward human tolerances to acceleration, but it is felt that some of the data obtained may be of value in future development work. At the present time, it appears that the most likely approach to the solution of this problem lies in the deformation and/or shear of metallic materials. However, the gas filled bag has possibilities provided out-gassing after impact can be eliminated.

H. PROBLEM AREAS AND WORK SCHEDULE

1. Problem Areas

It can be expected that many minor problem areas will develop as the development of the lunar roving vehicle progresses. However, no insurmountable problems can be foreseen for the proposed vehicle. The major anticipated problem areas and the proposed solutions based on present knowledge and experience are listed in Table I.

2. Priority and Work Schedule

Complete cooperation and coordination of efforts between the various agencies concerned with development of the roving vehicle will be necessary in order to establish and attain an acceptable target date for completion. A Tentative Priority and Work Schedule for the Roving Vehicle is given in Table II.

PROBLEM AREAS IN DEVELOPMENT OF THE LUNAR ROVING VEHICLE

COMPONENT	PROBLEM	PROPOSED SOLUTION
Turbine Bearing	Non-lubricated bearing capable of operating in mercury vapor atmosphere at 1200° F.	Graphite or molybdenum disulfide journal bearings.
Alternator Bearing	Lubricated or non-lubricated bearings capable of operating at 300° F in a vacuum.	Lubricated: Use low vapor pressure lubricant such as "Convoil-20" *(10^{-3} mm Hg @ 100° C) in conjunction with seal to limit leakage to a tolerable degree and ball or roller bearings. Non-lubricated: Same as turbine.
Wheel and payload pivot bearings	Lubricated bearings capable of operating in a vacuum at 300° F.	Same as lubricated alternator.
Deceleration Device	Develop decelerator capable of absorbing energy of impact at constant rate of deceleration and not exceed the imposed weight limitation.	Utilize energy required to shear metal.
Seals	Develop efficient low friction seals to operate in a vacuum and at temperatures ranging from 300 - 500° F.	Labyrinth type for turbine and lip seals or carbon ring seals for wheels, alternator and payload bearings.

*High Vacuum Vapor Pumps - Rochester Division Consolidated Electrodynamics Bulletin 6-1, Vanvary 1958

I. GROUND TEST FACILITIES

1. General

In order to meet the work schedule as proposed in Table II (Preceeding Chapter) the facilities required for lunar environmental simulation testing must be available by 1 July 1961. A complete study is presently being made to determine the requirement for additional facilities at ABMA necessary to accomplish this testing. It appears now that a large vacuum test chamber will be the major additional facility required. This requirement and proposed solution is discussed briefly below. More details on the vacuum chamber requirements and other requirements will be included in the complete Ground Test Facility Requirements Report now being prepared.

2. Vacuum Facility Requirements

General requirements for the vacuum test facility are listed below.

a. A vacuum system large enough to enclose a vehicle 24' x 20' x 24' high at pressures of 10^{-5} mm Hg or below. Incorporated within this chamber should be:

(1) Moveable overhead solar radiation spectrum capable of 420 BTU/hr/ft^2 intensity. This solar radiation source should be constructed so that the emitted rays are parallel to one another and completely encompass the vehicle.

(2) A heat sink of LN_2, or some other suitable cryogenic fluid to simulate outer space, should be located along the walls and portions of the ceiling.

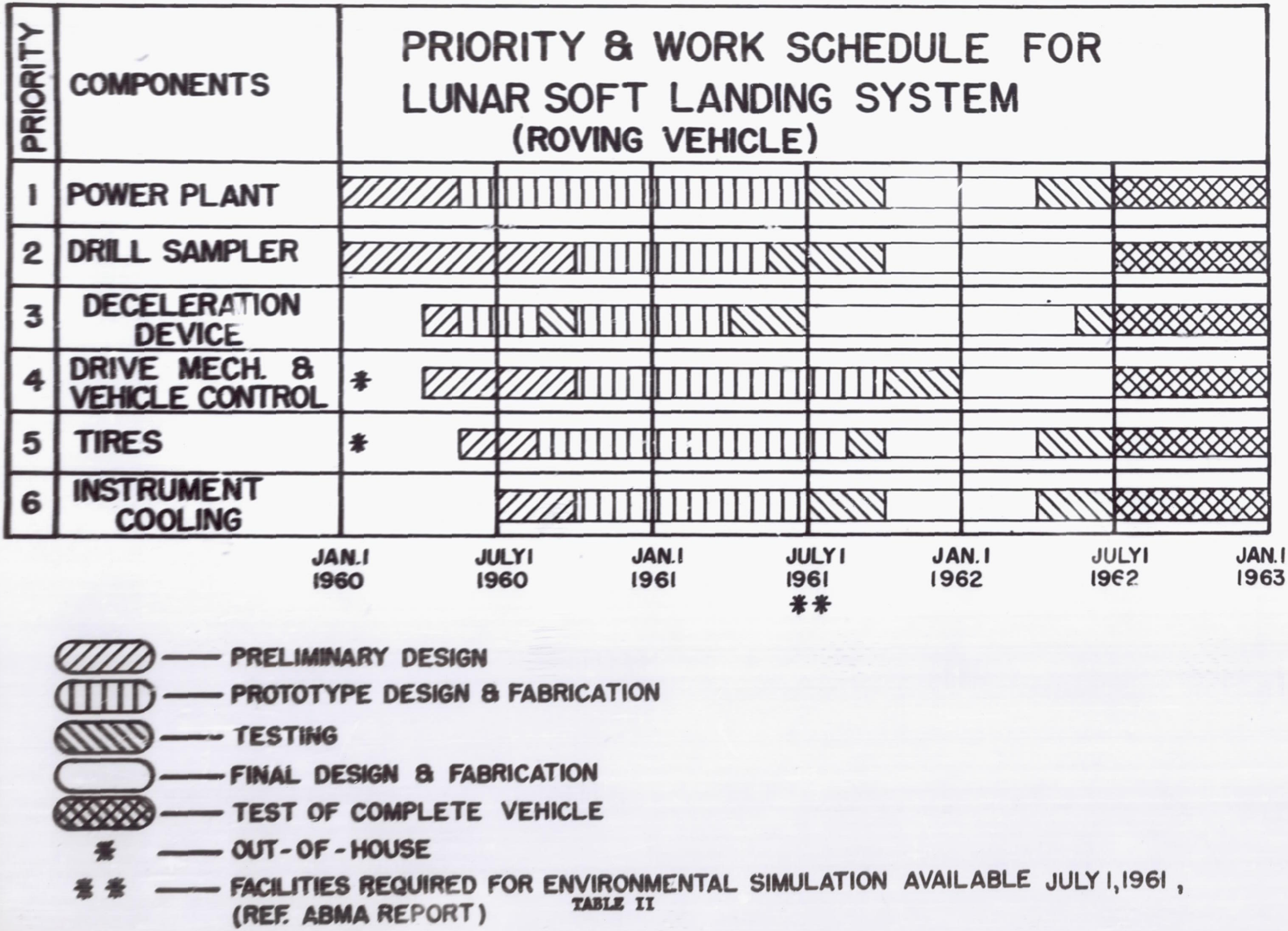

PRIORITY & WORK SCHEDULE FOR
LUNAR SOFT LANDING SYSTEM
(ROVING VEHICLE)
PRIORITY
COMPONENTS
1 POWER PLANT
2 DRILL SAMPLER
3 DECELERATION DEVICE
4 DRIVE MECH. & VEHICLE CONTROL *
5 TIRES *
6 INSTRUMENT COOLING
JAN. 1 1960
JULY 1 1960
JAN. 1 1961
JULY 1 1961 **
JAN. 1 1962
JULY 1 1962
JAN. 1 1963
PRELIMINARY DESIGN
PROTOTYPE DESIGN & FABRICATION
TESTING
FINAL DESIGN & FABRICATION
TEST OF COMPLETE VEHICLE
* — OUT-OF-HOUSE
** — FACILITIES REQUIRED FOR ENVIRONMENTAL SIMULATION AVAILABLE JULY 1, 1961,
(REF. ABMA REPORT)

TABLE II

(3) An infrared radiation source covering the chamber floor and lower walls to simulate radiation from the lunar surface.

(4) Instrumentation for all required measurements (temperature, pressure, etc.) on the vehicle and throughout the chamber.

(5) Provision for simulation of lunar surface conditions (dust, etc).

(6) Provision for impact testing of shock absorber.

b. A relatively small vacuum chamber, for example (6' dia. x 6' long), capable of 10^{-9} to 10^{-11} mm Hg for component testing. Incorporated within this chamber should be an ultra-violet and infrared radiation source.

3. Existing Facilities at ABMA

Vacuum test facilities now available at ABMA are located in Buildings 4619 and 4750. These facilities are enumerated and described below:

a. Building 4619

(1) Cylindrical Chamber. This chamber is four feet in diameter and six feet long. It is equipped with a 32" diffusion pump having a pumping speed of 15,000 liters per second. This pump is backed by an oil ejector pump which is in turn backed by a Kinney pump. A second Kinney pump is used for roughing. The diffusion pump has a "chevron" trap on the intake (upper) side which is filled with LN_2 to distill the oil which might otherwise go into the tank. This cuts the pumping speed about 40% but will permit a lower pressure to be obtained in the tank. Above the chevron trap is a right angle valve which also reduces pumping speed but enables the tank to be let down

to dry air without affecting the diffusion pump. The tank is assembled directly to the right angle valve. The ejector pump and the Kinney pumps are located below the tank on the floor level. The tank is about 10 feet above the floor. The 32" diffusion pump requires about two hours warm-up time after which the tank can be reduced to 2 - 5 x 10^{-6} mm Hg in about one hour. A low empty tank pressure of 5 x 10-7 (3.8 x 10^{-10} atm) has been reached after about one day of pumping.

A proposal for a change to increase flexibility is now being processed. For this change the present tank will be replaced with a manifold approaching a cube in shape with a 48" opening on each face. The diffusion pump would be attached to the bottom with a straight-through valve. The top and the four side ports would be available for special tanks as needed or be blanked off. The 48" bottom port would allow a 48" diffusion pump (which has just recently become available) to replace the present 32" pump, allowing a larger tank to be used. The new system should be available about 1 December 1959.

(2) <u>Tenny Walk-In Chamber</u>. This is a 7' x 8' x 10' tank with steel lining and insulation between the lining and the tank walls. Low pressure is about 300 microns (2¼ x 10^{-4} atm) and temperature environments from - 100°F to +300°F can be created. Only mechanical pumps are used.

(3) <u>Tenny Explosive Chamber</u>. This unit is four feet in diameter and five feet long. Pressure equivalent to 80,000 feet altitude is obtainable. The same pumping system is used for this chamber and the walk-in chamber.

(4) Special Chamber. Special studies are carried out in another chamber which is approximately a four foot cube in shape.

b. Building 4750

There are several large vacuum tanks in this building. One is a long cylindrical tank having a volume of 42,000 cu. ft. with an opening at the end about six feet in diameter. The inside diameter is about ten feet. The tank is vented to some large stacks which have steam ejectors at the top for removal of large quantities of gas. These are not presently being used. Another tank of similar shape but slightly smaller is attached to the same pumping system. Both of these tanks have been used for missile testing operations and are too dirty for "clean" use. The low pressure capability of these tanks is equivalent to approximately 100,000 ft. altitude or about 4 mm of mercury. There is a third tank which is quite clean and is painted white on the inside. It is a cylindrical tank 12 feet in diameter and has a volume of about 9,000 cu. ft. All of the tanks are piped to the same pumping system, which consists of two Fuller pumps driven by a 200 hp motor. The low pressure capacity of these pumps according to their name-plate is 29.7" of vacuum, which corresponds to 5.4 mm of mercury pressure. This appears conservative since lower pressures have been reported. The Fuller pump has an eccentric rotor with sliding vanes which are forced outward by centrifugal force and wipe the walls of the cylinder taking out the air. These are highly efficient for moving large quantities of air but are not capable of pumping low pressures because there is only line contact between the vanes and the cylinder walls.

4. Proposed Vacuum Facilities

Since none of the vacilities enumerated above meet the lunar vehicle testing requirements it is proposed that such a facility be constructed at ABMA. Details on this facility cannot be stated at present. However, the large chamber should include a full size end opening to permit passage of the lunar vehicle and test equipment. A personnel air lock and observation windows must also be provided. Since the extremely low pressures cannot be approached in large chambers with presently available equipment it is proposed that most of the system tests be conducted at pressures in the order of 10^{-5} mm Hg. However, the small component tests and studies of the effects of near vacuum conditions on such items as low vapor pressure liquid lubricants, out-gassing of materials, and bearing friction will be conducted at much lower pressures in a small chamber at pressures in the order of 10^{-9} to 10^{-11} mm Hg. Selection of the pumping system components will depend upon the time that may be allowed for pump-down, the out-gassing of materials in the vehicle and equipment, and the required minimum pressure. The chamber should be provided with facilities for creating the wide range of temperature conditions expected during the lunar day and night. Sufficient heating elements to produce 420 BTU per square foot per hour in the solar spectrum over the floor area will be placed in the upper portion of the chamber to simulate the radiant heat from the sun. The upper surface and sides of the chamber will be covered with plates containing tubing through which LN_2 is ciruclated to create a heat sink simulating outer space. The plates and tubes will be treated for high emissivity on the inner

side and high reflectivity on the side next to the chamber. They will be supported by stainless steel wires for low conductivity.

Further details on this proposed facility will be forthcoming in the report mentioned above.

APPENDIX I

INSTRUMENT COMPARTMENT COOLING SYSTEM CALCULATIONS

1. General Assumptions

a. Continuous internal heat load from instruments 100 watts (5.688 BTU/min).

b. Instrument compartment size 3' x 2' x 4' = 52 sq. ft. of surface area.

c. Linde S-14 type insulation to be used.

d. Weight of compartment 300 lbs.

The instrument compartment will contain only those instruments which will not operate under lunar environmental conditions. The instrument compartment will be a separate insulated temperature controlled compartment within the payload package. Either structural members or the insulating material may be used to support the compartment within the payload package. Regardless of which method of support is used the prime considerations are strength and minimum heat transfer into the compartment. An example of a design using structural members to support the instrument compartment would be to suspend the compartment on piano wires within the payload package. Since the use of metallic supporting members is not good from a heat transfer standpoint, the use of insulation material for support must be considered. The Linde S-14 insulation has a heat transfer coefficient $K = 2.5 \times 10^{-5}$ BTU/sq.ft./hr/°F under no load conditions. Under a 15 psi bearing load $K = 100 \times 10^{-5}$. The maximum total load on any supporting system would occur under landing conditions with 20g deceleration. ($F = {}^{w}/g \times 20g = 300 \times 20 = 6000$ lbs). If it were assumed that the insulation on the small end of the compartment (862 in^2) experienced this full load the resulting

pressure would be 7 psi.

2. Typical Load Calculations

The following calculations were made to determine which type of support would allow the least heat transfer into the compartment.

PIANO WIRE SUPPORTS: For the purpose of these calculations it is assumed that the compartment is designed to withstand 20 "G" deceleration in all directions. The following assumptions are made concerning piano wire and insulation.

Piano wire stressed to 100,000 psi

$K = 2.5 \times 10^{-5}$ BTU/sq.ft./hr/°F (Linde S-14 insulation)

Average inside compartment temperature 50°F

Outside temperature 250°F

Average wire temperature 50°F

Wire length L = 6 in. (0.5 ft.)

Number of wires 24

Emissivity of wire E = .1

Thermal conductivity of wire = 20 BTU/hr/ft/°F

α (Stefan - Boltzmann's constant) = $.173 \times 10^{-8}$ BTU/hr/sq.ft./°R^4

Amount of Wire Needed in Cross Sectional Area

$$\text{Area} = \frac{6000}{100{,}000} = .06 \text{ in}^2$$

Total cross sectional area needed for support in 6 different directions will be $6 \times .06 = .36$ in^2. If four wires are used for each direction of support their diameter will be .138 in. (d)

Total heat transfer = transfer through insulation + conduction down wire + radiation to wire.

Heat Transfer through insulation (Q_I)

$Q_I = KA \Delta T = 2.5 \times 10^{-5} \times 52 \times (250 - 50) = .26$ BTU/hr

Conduction Down Wire (Q_c)

$Q_c = KA\frac{\Delta T}{L} = (20)\ \frac{(.36)}{144}\ \frac{(200)}{(.5\ )} = 20$ BTU/hr

Radiation to Wire (Q_r)

Area of wire exposed to radiation (A)

$A = \pi\ dL = \frac{(.138)}{12}(.5)\ (24) = .433\ ft^2$

$Q_R = \sigma AE\ (T_1{}^4 - T_2{}^4) = .173 \times 10^{-8} \times .433 \times .1\ (710^4 - 510^4)$

$Q_R = 14.0$ BTU/hr

Total Heat Transfer

$Q = Q_I + Q_c + Q_R = .26 + 20.0 + 14.0 = 34.26$ BTU/hr

Heat Transfer Using Insulating Material as Support

Assume insulation subjected to landing loads from any direction. Even though the expected pressure is only 7 psi, due to lack of other data the (K) value of 100×10^{-5} @ 15 psi will be used.

$Q = KA \Delta T = 100 \times 10^{-5} \times 52 \times 200 = 10,4$ BTU/hr or 0.174 BTU/min.

Based on these calculations it is proposed that the insulation serve as the instrument compartment supporting system. There are also two other sources of heat transfer which must be considered. Provision must be made to circulate the cooling medium and pass electrical lines through the compartment wall. Since the cooling ducts will also be insulated they will not influence the heat transfer greatly and may be disregarded. However, the electrical cables will be the source of some heat transfer. For the purpose of calculations the following assumptions are made:

a. All wires are insulated so that heat transfer is limited to conduction down the wire.

b. The wires are copper with $K = 215$ BTU/hr/ft/°F.

c. Average wire length $X = 4$ ft.

d. Total cross sectional area of copper = .75 in dia. circle.

e. Assuming wire cold end temperature to be 140°F by locating outlet at high temperature part of the compartment.

f. Hot end temperature 250°F.

<u>Heat Transfer</u> (Q)

$$Q = KA\frac{\Delta T}{X} = (215)\,\frac{(.785)\quad(.75)^2}{144}\,\frac{(250-140)}{4} = 17/\text{BTU/hr or } .275 \text{ BTU/min.}$$

This additional heat transfer is common to both the structurally supported and insulation supported compartment and does not effect the previously made choice.

<u>Total Heat Load on Cooling System is</u>:

Internal load - 100 watts or	5.688 BTU/Min
Load through insulation	0.174
Load through electric wires	.275
Total	6.137 BTU/Min.

3. Power Requirements

ASSUME: COMPRESSOR MECHANICAL EFF. - 70% -
POLYTROPIC COMPRESSOR - N= 1.35
ADIABATIC EXPANSION
EFF. RESULTING FROM FLUID FRICTION LOSSES - 85% -
MECHANICAL EFF. - 90% -
5 PSIA INST. COMPARTMENT PRESSURE
COMPRESSION RATIO 6 TO 1
HYDROGEN TEMP. LEAVING INST. COMP. - 140°F

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} ; \quad T_2 = (600)(6)^{\frac{1.35-1}{1.35}} = (600)6^{.259} = (600)(1.59) = \underline{954°R}$$

$$\frac{T_{4S}}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{K-1}{K}} ; \quad T_{4S} = (760)\left(\tfrac{1}{6}\right)^{\frac{.41}{1.41}} = (760)(.594) = 451°R$$

$$\eta = \frac{h_4 - h_3}{h_{4S} - h_3} = \frac{T_4 - T_3}{T_{4S} - T_3} ; \quad T_4 = T_3 - \eta(T_{4S} - T_3) = 760 - (.85)(760 - 451)$$

$$T_4 = 498°R$$

HYDROGEN FLOW RATE (W)

$$W = \frac{Q}{\Delta T C_p} \qquad Q = \text{HEAT LOAD} = 6.137$$

$$W = \frac{6.137}{(600-498)(3.42)} = .0176 \ \frac{\#}{MIN}$$

COMPRESSOR H.P.

$$HP = \left(\frac{N}{N-1}\right)\frac{RT}{778}\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]\frac{W}{42.45} = \left(\frac{1.35}{.35}\right)\frac{(767)(600)}{(778)}\left[6^{.259} - 1\right]\frac{.0176}{42.45}$$

$$HP = \frac{.556}{\eta_M} = \underline{.795}$$

EXPANDER OUTPUT

$$HF = \left(\frac{K}{K-1}\right)\frac{RT_3W}{(778)42.45}\left[\left(\frac{P_4}{P_3}\right)^{\frac{K-1}{K}} - 1\right]\eta_F\eta_M = \underline{.326} \text{ H.P.}$$

NET H.P. REQ'D.

.795
.326
.469 H.P. INPUT FROM MOTOR

ACKNOWLEDGEMENTS

The many valuable contributions to this report by personnel of Systems Support Equipment Laboratory and others are hereby gratefully acknowledged.

Ground Support Equipment	Mr. Warren G. Hunter SATURN Program Coordinator, SSEL
Manned Lunar Capsule Recovery	Mr. Julian S. Hamilton Ch, Trans & Pack Sect., SSEL
Lunar Roving Vehicle	Mr. W. E. Dempster, SSEL Mr. J. R. Oliver, SSEL Mr. W. J. Higgins, SSEL Mr. J. E. Baggenstoss, SSEL Mr. R. F. Gentile, SSEL Mr. T. J. Lee, SSEL Mr. F. C. Eddens, Materials Br., S&M Lab

APPROVED: Report No. DLS-TN-26-59

Georg von Tiesenhausen Theodor A. Poppel

GEORG VON TIESENHAUSEN
Asst Ch, Mechanical Branch

for HANS HUETER
Dir, Systems Support Equip Lab

INTERNAL DISTRIBUTION
- ORDAB-D
  - Director
  - Dr Lange
- ORDAB-DS
  - Director
  - Mr Barker
- ORDAB-DV
  - Director
  - Mr Bucher
- ORDAB-DL
  - Director
  - Mr von Tiesenhausen
  - File

SECOND EDITION DISTRIBUTION

DISTRIBUTION
- ORDAB-X
- ORDAB-D
  - Dr von Braun
  - Dr Reese
- ORDAB-DA (2)
- ORDAB-DAS
- ORDAB-DAP
- ORDAB-DG (2)
- ORDAB-DGT
- ORDAB-DGE
- ORDAB-DM
- ORDAB-DV
- ORDAB-DS (2)
- ORDAB-DSP (2)
- ORDAB-DR
- ORDAB-DL
- ORDAB-DL/Tech & Scientific Staff
- ORDAB-DLE (2)
- ORDAB-DLM (4)
- ORDAB-DLS (2)
- ORDAB-DT
- ORDAB-HT (8)
- ORDAB-DL/Files (3)